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STEAM-TURBINE PRINCIPLES AND PRACTICE

TERRELL CROFT, EDITOR

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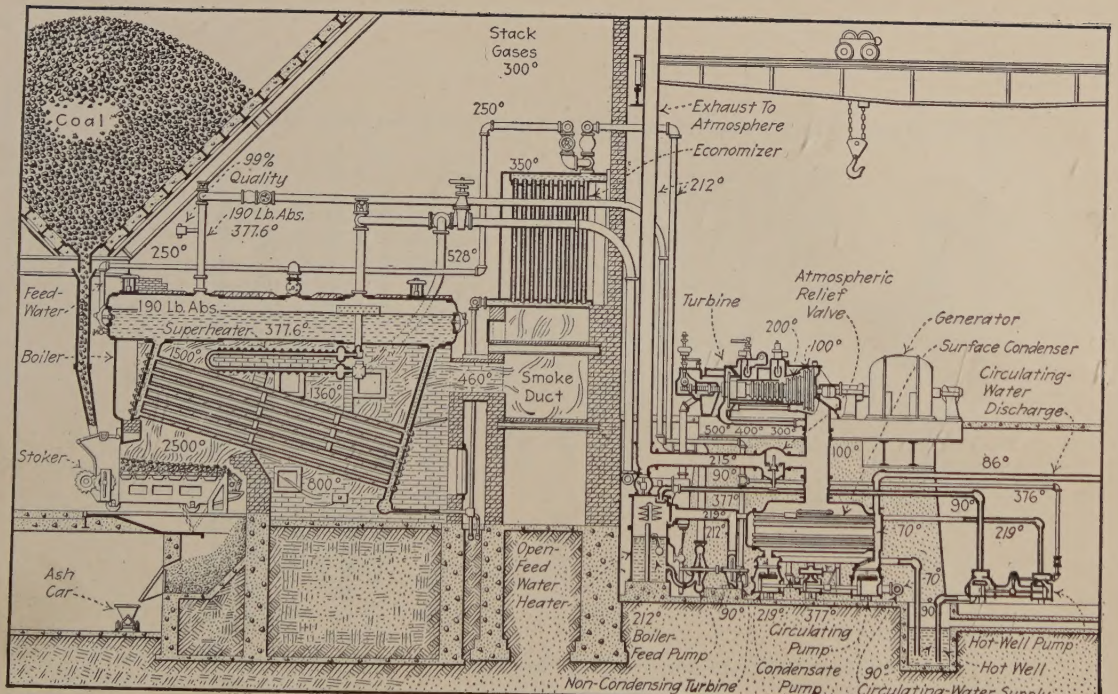

Sectional view of the apparatus and piping in a typical turbo-generator power station which is operating condensing. (This shows the variation of temperature and pressure of the furnace gases and of the steam during its course from the boiler through the turbine, condenser and auxiliaries and back to the boiler. From supplement to *Power*, March 21, 1916.)

Frontispiece.

STEAM-TURBINE PRINCIPLES AND PRACTICE

TERRELL CROFT, Editor

CONSULTING ENGINEER. DIRECTING ENGINEER, TERRELL CROFT ENGINEERING CO.
MEMBER OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.
MEMBER OF AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS.
MEMBER OF THE ILLUMINATING ENGINEERING SOCIETY.
MEMBER AMERICAN SOCIETY TESTING MATERIALS.

First Edition

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PREFACE

Although the steam turbine is a relatively new development in steam power-plant practice, it is already of great importance. Its adoption has, because of its economic superiority for many conditions, been very rapid. Today, turbines of different capacities ranging from 1 hp. up to 80,000 hp. are being effectively utilized for power generation. The number of turbines in use will soon exceed—if it does not already exceed—the number of reciprocating steam engines. It follows that all *successful* power-plant men must *now* be informed concerning these machines.

STEAM-TURBINE PRINCIPLES AND PRACTICE has been prepared, for the "practical" man, to furnish this information. It has been written to provide the operating engineer, the plant superintendent, or manager with such steam-turbine information as he requires in his everyday work. The aim has been to treat only topics of two general classes: (1) *Those with which a man must be familiar to insure the successful and economical operation of steam turbines.* (2) *Those a knowledge of which is necessary to enable a man—one who is not familiar with the details of its design or theory—to make a wise choice if he contemplates the purchase of a turbine.* Only sufficient theory is given to insure a sound understanding of the principles of turbine operation. The "design" of turbines is not treated at all. A working knowledge of arithmetic will enable one to read the book intelligently.

Drawings for nearly all of the 282 illustrations were made especially for this work. It has been the endeavor to so design and render these pictures that they will convey the desired information with a minimum of supplementary discussion.

Throughout the text, principles which are presented are explained with descriptive expositions or worked-out arithmetical examples. At the end of each of the 14 divisions there are questions to be answered and, where justified, problems to

be solved by the reader. These questions and problems are based on the text matter in the division which they follow. If the reader can answer the questions and solve the problems, he then must be conversant with the subject matter of the division. Detail solutions to all of the problems are printed in the appendix in the back of the book.

As to the method of treatment: Fundamental principles of turbine operation are first presented so as to provide a knowledge of the theory which is necessary for the understanding of how turbines function. This is followed by a division on turbine classifications and nomenclature. Next, the principal turbine types and constructions are described and discussed. Then follows a division on steam-turbine installation. This is followed by four divisions on important turbine parts which require periodic attention. These divisions treat of: Shafts, bearings, and packing glands; governors and valves; reduction gears and couplings; regenerators and condensers. Next is an important division on high-pressure, bleeder, mixed-pressure, and exhaust-steam turbines.

Following this are practically-treated divisions on lubrication and operation and maintenance. The next division on testing explains the methods whereby the efficiencies of turbines are determined. The last two divisions—one on the effects of steam pressure, superheat, and vacuum on steam-turbine economy; the other on steam-turbine economics and selection—outline the processes by which the most economical steam conditions and the most economical turbine may be selected for a given installation. They also interpret the effects of steam-condition changes on the economy of the turbine.

With this, as with other books which have been prepared by the editor, it is the sincere desire to render it of maximum usefulness to the reader. It is the intention to improve the book each time it is revised and to enlarge it as conditions may demand. If these things are to be accomplished most effectively, it is essential that the readers coöperate with us. This they may do by advising the editor of any alterations which they feel it would be advisable to make. Future revisions and additions will, insofar as is feasible, be based on such suggestions and criticisms from the readers.

Although the proofs have been read and checked very carefully, it is possible that some undiscovered errors may remain. Readers will confer a favor in advising the editor of any such.

TERRELL CROFT.

UNIVERSITY CITY,
ST. LOUIS, MO.,
December, 1922.

ACKNOWLEDGMENTS

The editor desires to acknowledge the assistance which has been rendered by a number of concerns and individuals in the preparation of this book.

Portions of the text material appeared originally as articles by the editor in certain trade and technical periodicals among which are *Power*, *Power Plant Engineering*, and *Southern Engineer*. In all such cases and in others where material from publications has been used, it is believed that proper acknowledgment has been accorded at the proper place in the text.

The list of manufacturers who coöperated in supplying text data and illustrations would include practically all manufacturers of steam turbines, both large and small, in the United States. In virtually all cases where such data have been used, special acknowledgement is accorded in the text. The editor is particularly indebted to the *Allis-Chalmers Manufacturing Company* and the *Terry Turbine Company* for the data which was submitted by their chief turbine engineers.

Assistance and information have been obtained from certain recognized technical turbine books and, in some cases, tables and other data were taken from them. Proper acknowledgment is accorded in the text wherever such data were used.

Special acknowledgment is hereby accorded Edmond Siroky, Head Mechanical Engineer of the Terrell Croft Engineering Company, who has been responsible for the technical accuracy of the book.

Other acknowledgments have been made throughout the book. If any has been omitted, it has been through oversight and, if brought to the editor's attention, it will be incorporated in the next edition.

TERRELL CROFT.

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STEAM-TURBINE PRINCIPLES AND PRACTICE

DIVISION 1

STEAM-TURBINE FUNDAMENTAL PRINCIPLES

1. A Steam Turbine Operates By Virtue Of Heat which it abstracts from the steam and which it converts into mechanical work. Heat and mechanical work are both forms of energy (see the author's PRACTICAL HEAT) and can, therefore, be converted one to the other by the proper means. Any apparatus which can convert heat energy into mechanical work is called a *heat engine*. Thus, the steam turbine is just as much a heat engine as is a steam or internal-combustion engine. The steam turbine is different, however, in the manner in which it converts the heat energy into mechanical work, as will be explained.

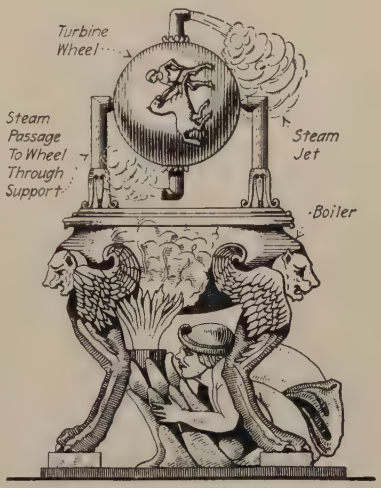


FIG. 1.—The earliest known heat engine, described by Hero of Alexandria about 150 B. C., was a reaction turbine.

NOTE.—THE STEAM TURBINE WAS THE FIRST FORM OF HEAT ENGINE TO BE DEVELOPED AND THE LATEST TO BE PERFECTED, thus it might be said that it is the oldest as well as the newest form of heat engine. The earliest record of any heat engine is in a book written by Hero of Alexandria, probably about 150 B.C., in which a steam reaction wheel (Fig. 1) is mentioned. The next development on record was the turbine of

Branca, an Italian architect, (Fig. 2). These turbines will be described later.

The first turbine patent in the United States was issued in 1831 and under it several turbines were built. They seemed to give satisfaction for some time but they did not last long.

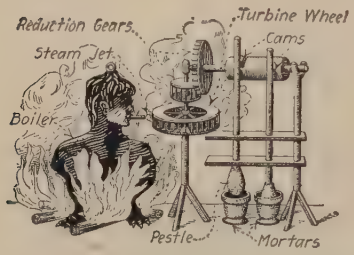


Fig. 2.—Branca's impulse turbine (1629).

The commercially successful turbine is due, however, to the inventive genius and experiments of De Laval and Parsons, who worked separately and along different lines, during the years 1882 to 1889. Turbines of large capacities have been developed only within the last twenty years.

2. In A Steam Turbine, Heat Energy Is First Converted Into Velocity Energy Or Kinetic Energy which is then converted into mechanical work. The fact that steam will issue with considerable velocity through any small opening in a containing vessel is, no doubt, known to all. It will be shown that the velocity is derived from heat energy which the steam liberates as it passes through the opening. It will also be shown that the velocity of the issuing steam can be forecasted with reasonable accuracy. Such calculations are, however, the work of the turbine designers and, except in so far as they explain fundamental principles of turbine operation, will not be treated herein.

3. A Steam Turbine Does Mechanical Work By Virtue Of The Velocity With Which The Steam Strikes Or Leaves Moving Parts.—As the steam attains its velocity, by issuing from an opening, its velocity energy may be converted into mechanical work by suitably deflecting its current. In this respect, a steam jet acts just as does a water jet. The manner in which fluid jets may produce forces will now be shown.

4. The Terms "Impulse" and "Reaction" Have Specific Meanings In Turbine-engineering Parlance.—These specific meanings, which are employed in this book and which are explained in the following sections, are different from the meanings of the same words as they are employed in physics, mechanics and in ordinary usage.

NOTE.—"THERE IS LITTLE CONNECTION BETWEEN THE USUAL MEANINGS OF THE WORDS 'IMPULSE' AND 'REACTION' AND THE SPECIFIC IDEAS THEY ARE INTENDED TO CONVEY IN STEAM-TURBINE PARLANCE. Actually, all commercial steam turbines work by both impulse and reaction. A German writer has used instead of 'impulse' and 'reaction' words meaning 'equal pressure' and 'unequal pressure,' which to the author seem much more appropriate." This paragraph abstracted from Moyer's STEAM TURBINES, *John Wiley & Sons.*

5. An Impulsive Force Or "Impulse" Is That Force Which Is Produced On An Object When A Fluid Jet Strikes The Object,

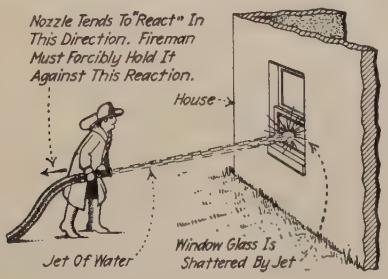

Nozzle Tends To "React" In This Direction. Fireman Must Forcibly Hold It Against This Reaction.
House
Jet Of Water
Window Glass Is Shattered By Jet

FIG. 3.—Illustrating the "impulse" effect of a jet of water when directed against a stationary object (window pane).

Fig. 3.—This is the specific turbine-engineering definition; see Sec. 4. The fact that a stream of water from a fire hose can shatter a pane of glass (Fig. 3) or even move heavier

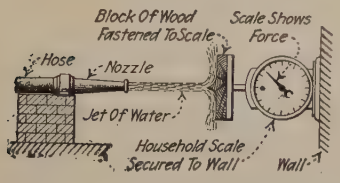


FIG. 4.—Measuring the "impulse" of a jet of water.

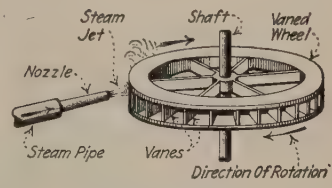


FIG. 5.—The elements of Branca's steam turbine (1629).

objects which it strikes is well known. A distinguishing characteristic of an impulsive force is that the fluid stream which strikes the object, and thereby produces the force, leaves the object at the same or at a less velocity than that with which it strikes the object. A simple means of measuring the impulsive force is shown in Fig. 4. The force which the

scale indicates is that which is produced by the impulse of the water jet.

Note.—This "Impulse" Principle Was Employed By Branca In His Primitive Turbine, Figs. 2 and 5, which was formed by mounting a number of vanes on the circumference of a wheel and directing a steam jet against them at one point. As the wheel is moved by the impulsive force, the steam jet plays successively on other vanes, thus providing a continuous motion of the wheel whereby it may be caused to do work. Those turbines which employ the impulse principle mainly for their operation are called *impulse turbines;* see Sec. 30.

6. A Large Impulsive Force Is Produced When A Fluid Stream Strikes An Object And Then Leaves It In An Opposite

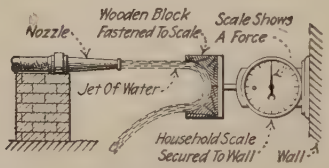


Fig. 6.—Measuring the impulse of a jet of water. (Compare with Fig. 4.)

Direction.—This may well be explained by a simple experiment, Fig. 6. When the fluid stream strikes an object which is so shaped that it reverses the direction of the stream, a

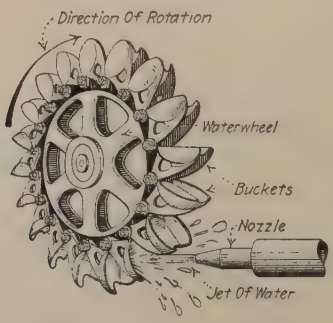


Fig. 7.—A Pelton waterwheel.

much greater impulsive force is produced than when the direction of the stream is not reversed (Fig. 4). This occurs in spite of the fact that the stream may leave the object with the same velocity as that with which it approached the object.

In fact, it is found that the force in Fig. 6 is just twice that of Fig. 4. (In ordinary parlance—not in steam-turbine parlance—the force produced by the jet *leaving* the object would be called a "repulsive" force.)

Note.—The Principle Of Thus Utilizing Large Impulsive Forces Is Employed In The Pelton Waterwheel (Fig. 7) and in the

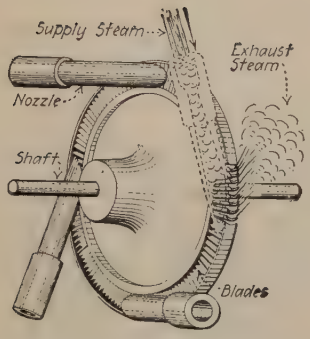


Fig. 8.—The De Laval trade mark which illustrates the principle of the so-called "impulse" turbine. (It utilizes impulsive forces.)

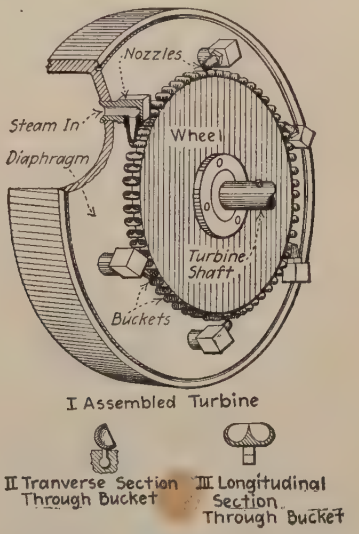


Fig. 9.—An early form of steam turbine which was patterned after the Pelton waterwheel.

De Laval single-stage turbine (Fig. 8). Some of the earlier turbines, as shown in Fig. 9, were made very similar to the Pelton waterwheel, but exactly this construction is no longer used (see Div. 3) because more efficient ones have been developed.

7. A Reactive Force Or "Reaction" Is That Force Which Is Produced On An Object When A Fluid Stream Leaves The Object At A Greater Velocity Than That With Which It Approaches The Object Figs. 1 and 10. This is the specific turbine-engineering definition; see Sec. 4. Examples of this force are, no doubt, familiar to everyone even though many people do not know their explanation. Some familiar

examples are: (1) *The "push-back" or "kick-back" of a hose,* as in Fig. 3, *or of a shot-gun.* (2) *The fireworks pin wheel,* Fig. 10–*I*. (3) *The revolving lawn sprinkler,* Fig. 10–*II*. The existence of a force when a fluid stream leaves a body may well be illustrated by the simple experiment, shown in Fig. 11 wherein the bucket is held from the vertical by the reactive force of the water jet. In each of these cases the velocity of the fluid stream which leaves the body is greater than that of the fluid stream approaching the body.

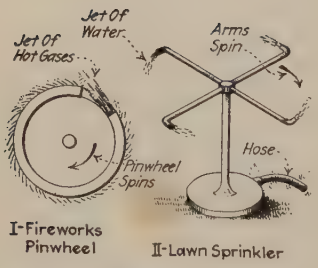


Fig. 10.—Illustrative examples of reaction wheels.

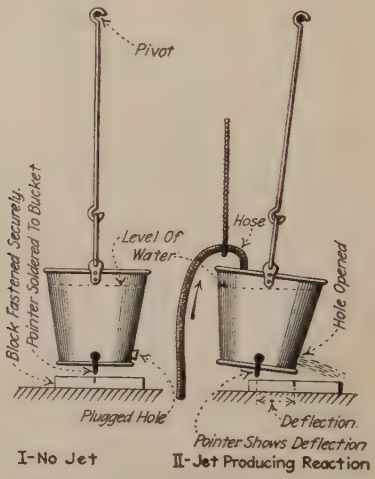


Fig. 11.—A simple experiment which proves the existence of a reaction force when a jet leaves an object. (The deflection is shown exaggerated for clearness.)

Note.—Reactive Forces, Which Were Produced At The Ends Of The Arms Of Hero's Turbine (Fig. 1) as steam issued from them, provided the rotational motion whereby work was done. Hero's turbine was, thus, similar to our common lawn sprinkler, Fig. 10–*II*. No modern turbines employ only reaction forces (see Sec. 31), but those turbines which employ the reaction principle mainly for their operation are called **reaction turbines.**

Explanation.—The Nature Of Reactive Forces can easily be understood by a study of Figs. 12, 13, and 14. Imagine a tank which has similar holes on opposite sides near its bottom, Fig. 12. If both holes are corked and the tank is filled with water, the water will exert a force on each cork tending to push it out. But, since the corks plug

equal holes and since both are subjected to the same pressure, the force on one cork is equal to that on the other. Say each cork is subjected to 2 lb. as in Fig. 12-*I*. If, now, one cork is removed as in *II*, then the one force of 2 lb. is removed and the other 2-lb. force must be balanced from

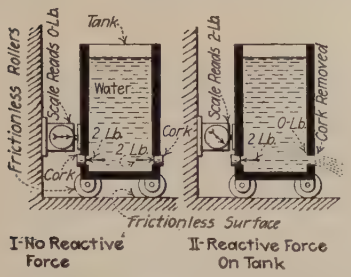


FIG. 12.—Illustrating how a reactive force comes into action. Note that the reactive force would exist even if the tank were in a vacuum.

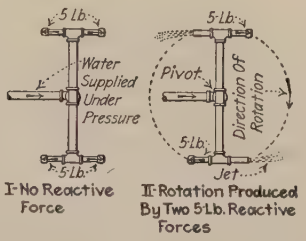


FIG. 13.—Showing the nature of the forces that cause rotation of a lawn sprinkler.

without the tank as shown. If the scale were not in a position to prevent it as shown, the remaining 2-lb. force in Fig. 12-*II* would be capable of moving the tank and thus doing work upon it.

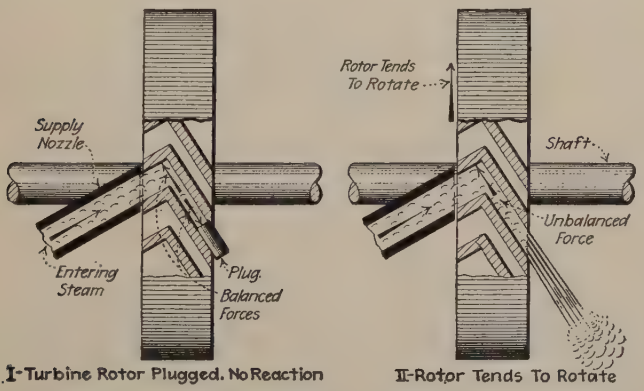


FIG. 14.—Illustrating the nature of reaction forces in a modern reaction turbine. The blades or nozzles are here shown with square corners merely to explain the principle. Actually, the nozzles of reaction turbines are formed with smooth curved surfaces as shown in other illustrations.

The mode in which reactive forces cause rotation is shown in Fig. 13. In *I* the pipes are capped at all ends and the forces are balanced as shown. When, however, the pipes are opened as in *II*, certain forces are removed and the remaining forces are free to turn the sprinkler.

In Fig. 14-*I* the rotor of a modern reaction turbine is diagrammatically shown with its outlet plugged. The internal forces on the plug and on the equal area at the inlet end of the nozzle are balanced as indicated. If, however, the plug is removed as in *II*, the force which it withstood is also removed and the force on the rotor blade is unbalanced. Hence, this force is capable of rotating the rotor.

8. Steam Liberates Heat As It Flows Through An Opening from a region of high pressure to one of lower pressure. Dry saturated steam at low pressures contains less heat (B.t.u.) per pound than does dry saturated steam at high pressures; a study of any steam table will confirm this statement, see also the author's PRACTICAL HEAT. Therefore, if dry saturated steam undergoes a reduction in pressure, it must either lose heat or become superheated. Experience shows, however, that when steam expands in a turbine nozzle the steam does not become superheated—in fact, it undergoes a reduction in quality or dryness. Therefore, in a turbine, the steam loses or liberates heat as its pressure is reduced. Experience further shows that steam, when it passes without friction through an opening, loses just as much heat as it would have lost, had it expanded adiabatically behind a piston from the same initial to the same final pressure. But, in flowing through a relatively small opening, the steam acquires a high velocity which it would not have acquired if it had expanded behind a piston. It can be conceived that, in flowing through an opening, the steam does its work upon the steam immediately ahead of it by pushing it forward so hard as to increase its velocity.

9. The Kinetic Energy Which Steam Acquires in flowing through an opening from a region of high pressure to one of lower pressure is equal (except for small losses) to the heat which is liberated by the steam. It is known that the work which steam does when expanding adiabatically behind a piston is equal to the heat that is liberated by the steam; see PRACTICAL HEAT. Hence, it follows, that the kinetic energy acquired by steam in flowing through an opening is equal to the work which the steam would have done if it had expanded adiabatically behind a piston. Obviously then, if a steam turbine could utilize all of the kinetic energy which its steam acquires, it would do exactly the same amount of work as

would a perfect steam engine which expanded steam adiabatically between the same pressures. The relation between heat energy and kinetic energy in a steam turbine is, therefore, neglecting small losses:

(1) *Kinetic energy acquired* = *Heat liberated by adiabatic expansion.*

10. The Velocity Theoretically Acquired By Steam In Flowing Through An Opening from a region of high pressure to a region of lower pressure may be readily computed if the state of the steam at the higher pressure is known and if the lower pressure is known. These known factors determine the amount of heat liberated by an adiabatic expansion. Hence, by For. (1), they also determine the kinetic energy acquired. The formula, (see also chart *C* of Fig. 15) which expresses the velocity acquired and which is derived below, is:

$$v = 223.7\sqrt{\mathbf{H}_1 - \mathbf{H}_2} \qquad \text{(ft. per sec.)} \tag{2}$$

Wherein: v = velocity acquired by the steam, in feet per second. $\mathbf{H}_1$ = total heat of the steam at the higher pressure, in British thermal units per pound; this may be obtained from a steam table or from the chart of Fig. 15. $\mathbf{H}_2$ = total heat of the steam after adiabatic expansion to the lower pressure, in British thermal units per pound; this may be obtained most readily from the chart of Fig. 15 as explained below.

Derivation.—From the mechanics of moving bodies, the kinetic energy of any moving body is:

$$\textit{Kinetic energy} = \frac{\mathbf{W}v^2}{2g} = \frac{\mathbf{W}v^2}{2\times 32.2} = \frac{\mathbf{W}v^2}{64.4} \qquad \text{(ft.-lb.)} \tag{3}$$

Wherein: $\mathbf{W}$ = the weight of the body, in pounds. v = the velocity of the body, in feet per second. g = the acceleration due to gravity, in feet per second per second (= 32.2).

Also, from the thermodynamics, see the author's PRACTICAL HEAT:

$$\textit{Heat liberated by adiabatic expansion} = \mathbf{W}(\mathbf{H}_1 - \mathbf{H}_2) \qquad \text{(B.t.u.)} \tag{4}$$

Or, since 1 *B.t.u.* = 778 *ft.-lb.*, expressing the same thing in foot-pounds:

$$\textit{Heat liberated by adiabatic expansion} = 778\mathbf{W}(\mathbf{H}_1 - \mathbf{H}_2) \qquad \text{(ft.-lb.)} \tag{5}$$

Wherein: $\mathbf{W}$ = the weight of the steam, in pounds. $\mathbf{H}_1$ and $\mathbf{H}_2$ have the meanings given above. 778 = the equivalent of 1 B.t.u. in foot-pounds.

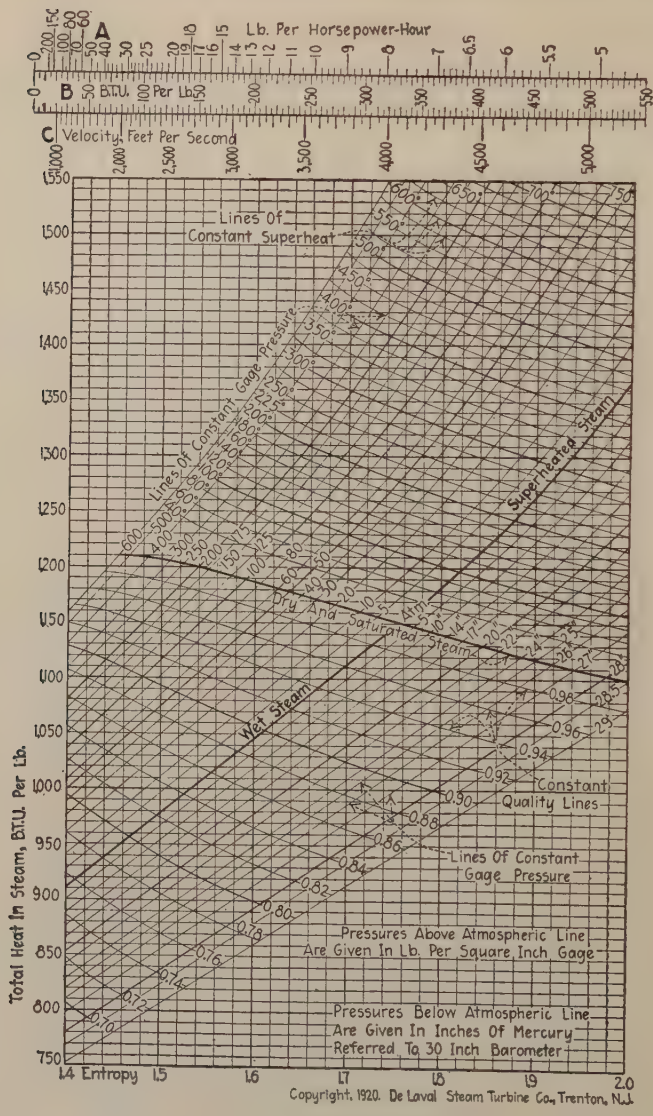

A Lb. Per Horsepower-Hour
B B.T.U. Per Lb.
C Velocity, Feet Per Second
Lines Of Constant Superheat
Lines Of Constant Gage Pressure
Superheated Steam
Dry And Saturated Steam
Wet Steam
Constant Quality Lines
Lines Of Constant Gage Pressure
Pressures Above Atmospheric Line Are Given In Lb. Per Square Inch Gage
Pressures Below Atmospheric Line Are Given In Inches Of Mercury Referred To 30 Inch Barometer
Total Heat In Steam, B.T.U. Per Lb.
1.4 Entropy 1.5 1.6 1.7 1.8 1.9 2.0
Copyright, 1920. De Laval Steam Turbine Co., Trenton, N.J.

Fig. 15.—Total-heat-entropy chart for steam.

Since steam, in flowing through an opening, may be assumed to possess no kinetic energy before it reaches the opening, For. (3) will express the kinetic energy which it acquires in passing through the opening. For. (1) states that:

(6) *Kinetic energy acquired = Heat liberated by adiabatic expansion*

Now, substituting values from Fors. (3) and (5) in For. (6):

$$\frac{\mathbf{W}v^2}{64.4} = 778\ \mathbf{W}\ (\mathbf{H}_1 - \mathbf{H}_2) \qquad \text{(ft.-lb.)} \tag{7}$$

Or, by transposing and simplifying:

$$v = 223.7\sqrt{\mathbf{H}_1 - \mathbf{H}_2} \qquad \text{(ft. per sec.)} \tag{8}$$

EXAMPLE.—What theoretical velocity will steam acquire when it is expanded through an opening from the dry saturated state at 200 lb. per sq. in. abs. to 15 lb. per sq. in. abs.? SOLUTION.—Use the chart of Fig. 15 to obtain the values for $\mathbf{H}_1$ and $\mathbf{H}_2$. In this chart, pressures above atmospheric are expressed as gage pressures. Now, 200 lb. per sq. in. abs. = 200 − 14.7 = 185.3 *lb. per sq. in. gage.* Also, 15 *lb. per sq. in. abs. = atmospheric pressure,* closely. Then, from Fig. 15 on the "Dry and Saturated Steam" line for 185.3 lb. per sq. in. gage, $\mathbf{H}_1 = 1198.1$ B.t.u. per lb. Also from Fig. 15, by adiabatic expansion—following vertically downward on Fig. 15 to 15 lb. per sq. in., $\mathbf{H}_2 = 1010$ B.t.u. per lb. Hence, by For. (2): $v = 223.7\sqrt{\mathbf{H}_1 - \mathbf{H}_2} = 223.7 \times \sqrt{188.1} = 223.7 \times 13.72 = 3{,}072$ *ft. per sec.*, which is about 15 miles per minute.

NOTE.—THE VELOCITY ACTUALLY ACQUIRED BY THE STEAM depends somewhat on the shape of the opening or *nozzle* through which the steam expands. As the steam flows through the nozzle, the friction of the steam on the walls of the nozzle produces heat which is returned to the steam and which raises the value of $\mathbf{H}_2$ in For. (2), thus reducing the amount of heat actually liberated by the steam. In a well designed nozzle, that is one in which friction has been minimized by properly shaping the nozzle, the actual velocity is usually about 95 per cent of the theoretical or computed by For. (2).

NOTE.—THE CALCULATION OF STEAM VELOCITIES WITH A TOTAL-HEAT-ENTROPY DIAGRAM, Fig. 15, is much simpler than by using the above formula. The entropy (see PRACTICAL HEAT) is the property of steam which does not change during an adiabatic expansion. Therefore, the heat liberation during an adiabatic expansion can be traced from a total-heat-entropy diagram by following along a vertical (constant entropy) line from a selected point representing the initial state of the steam to the line which represents the pressure into which the steam is being discharged. The heat liberation is read, as the difference between the initial B.t.u. value and the final B.t.u. value, along the vertical scale of the diagram. The velocity of the steam (theoretical) can then be read from the B.t.u. velocity scale, *C*, at the top of the main diagram, which was computed by using For. (2).

Example.—By using the chart of Fig. 15, determine the theoretical velocity with which steam, at 200 lb. per sq. in. gage and superheated 100° F., will issue from a nozzle into a region of 29 in. vacuum. Solution.—$\mathbf{H}_1$ is found, at the intersection of the 100° superheat line and the 200-lb. pressure line, to be 1259 B.t.u. per lb. Following vertically down to the 29-in. vacuum line and then to the left, $\mathbf{H}_2$ is found to be 863 B.t.u. per lb. Therefore, $\mathbf{H}_1 - \mathbf{H}_2 = 1259 - 863 = 396$ *B.t.u. per lb.* From the B.t.u. velocity scale, *C*, at the top, the *theoretical velocity* = 4,460 *ft. per sec.* (approximately 51 miles per minute).

Note.—The Chart Of Fig. 15 Is Drawn For Gage Pressures For Use In Districts Where The Average Barometric Pressure is about 30 in. of mercury column. Such charts are generally drawn (in other books) for absolute pressures; but it is believed that, for most purposes, one drawn for gage pressures (assuming an average atmospheric pressure) will be more useful for the operator. Hence, the one here given is so drawn. To use the chart in districts where the barometric pressure is much different than that specified above, or for extreme accuracy, the reader may calculate the pressure correction required for using this chart by the relations:

(9) *Gage pressure value to be used on chart = Actual gage reading −* [0.49 × (30 − *barometer reading*)]

(10) *Vacuum gage value to be used on chart = Observed vacuum gage reading +* (30 − *barometer reading*)

Or, one may employ a similar chart which is based on absolute pressures, for example Marks and Davis' Tables and Diagrams of the Thermal Properties of Saturated and Superheated Steam (Longmans, Green, and Co.).

11. The Velocity Of A Fluid Stream May Be Reduced As The Stream Passes Over A Moving Surface (Fig. 16). It is necessary to know something about this reduction of velocity in order that one may understand why turbine parts move at such great speeds and why staging (Sec. 17) is employed.

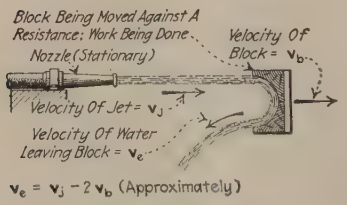

Block Being Moved Against A Resistance; Work Being Done
Nozzle (Stationary)
Velocity Of Block = v_b
Velocity Of Jet = v_j
Velocity Of Water Leaving Block = v_e
$v_e = v_j - 2v_b$ (Approximately)

Fig. 16.—Illustrating the loss of velocity by a stream of fluid as it does work.

Explanation.—In Fig. 6 it was shown how a fluid stream may produce a force but, since in Fig. 6 the force of the stream did not move the block and hence did no work, the velocity of the stream was undiminished (except for a slight loss due to friction) as it passed over the surface of the block. But if, now (Fig. 16), the block is free to move, and does

move away from the stationary nozzle with a velocity v_b, then obviously the stream will not *approach* the block with its full velocty v_j. The stream will only strike the moving block with a velocity equal to $v_j - v_b$; just as when an automobile going at 45 mi. per hr. overtakes a train traveling at 30 mi. per hr.; see also Fig. 17. Thus, if the velocity of the

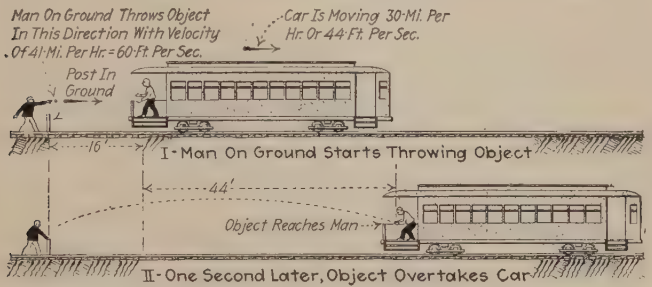

Man On Ground Throws Object In This Direction With Velocity Of 41 Mi. Per Hr. = 60 Ft. Per Sec.
Car Is Moving 30 Mi. Per Hr. Or 44 Ft. Per Sec.
Post In Ground
16'
I - Man On Ground Starts Throwing Object
44'
Object Reaches Man
II - One Second Later, Object Overtakes Car

FIG. 17.—Showing why one moving object strikes another only with the *difference* of their velocities. To the man on the platform the object seems to come with a velocity of only: $60 - 44 = 16$ *ft. per sec.* or 11 *mi. per hr.*

stream is 3,000 ft. per sec. (Fig. 18) and that of the moving block is 1,400 ft. per sec., the stream overtakes or approaches the block with a velocity of $3{,}000 - 1{,}400 = 1{,}600$ *ft. per sec.*

Now, in passing over the surface of the moving block, the stream passes just as fast over a point where it leaves the block as it passes over

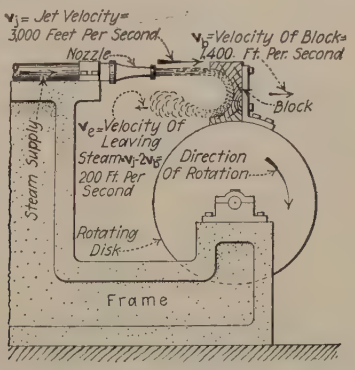

v_j = Jet Velocity = 3,000 Feet Per Second
Nozzle
v_b = Velocity Of Block = 1,400 Ft. Per Second
Block
v_e = Velocity Of Leaving Steam = $v_j - 2v_b$ = 200 Ft. Per Second
Direction Of Rotation
Steam Supply
Rotating Disk
Frame

FIG. 18.—Illustrating the velocities of a fluid stream as it strikes a moving surface turbine blade) and then leaves in an opposite direction.

a point where it strikes the block (except for a slight frictional loss). Therefore, the stream leaves the block with the same velocity as that with which it struck the block, or with a velocity of $v_j - v_b$ to the left. That is, in Fig. 18, the stream leaves the block with a velocity of 1,600 ft. per sec. (with relation to the moving block).

But, since the block is moving away from the nozzle to the right with a velocity of v_b, the actual velocity of the leaving steam jet toward the stationary nozzle is made up of—is the difference between—the velocity with which the stream passes from the block and the velocity of the block itself just as when one throws an object with a velocity of 41 mi. per hr. from the rear end of a train traveling at 30 mi. per hr., the object appears, to an observer on the ground, to move only at the rate of 11 mi. per hr. see Fig. 19. Therefore the stream (Fig. 16) actually leaves the block with a velocity toward the stationary nozzle of $v_e = (v_j - v_b) - v_b = v_j - 2v_b$. Thus, also in Fig. 18, the stream from the moving block approaches the stationary nozzle only with a velocity of 1,600 − 1,400 = 200 *ft. per sec.*

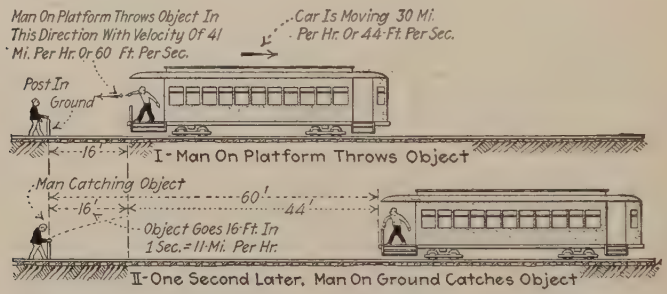

Man On Platform Throws Object In This Direction With Velocity Of 41 Mi. Per Hr. Or 60 Ft. Per Sec.
Car Is Moving 30 Mi. Per Hr. Or 44 Ft. Per Sec.
Post In Ground
16
I - Man On Platform Throws Object
Man Catching Object
60
16
44
Object Goes 16 Ft. In 1 Sec. = 11 Mi. Per Hr.
II - One Second Later, Man On Ground Catches Object

Fig. 19.—Showing why, when an object is discharged from a moving body, the actual velocity of the object is the difference between the velocity of discharge and the velocity of the moving body. To the man on the ground the object seems to come with a velocity of only 16 ft. per sec. or 11 miles per hour.

12. Kinetic Energy Is Converted Into Work As The Velocity Of A Jet Is Reduced in passing over a moving surface. The truth of this is evident by inspection of For. (3). Applying For. (3) to the example of Fig. 16 it follows that:

$$\text{(11)} \qquad \textit{Kinetic energy of jet} = \frac{\mathbf{W}v_j^2}{64.4} \qquad \text{(ft.-lb.)}$$

Wherein: $\mathbf{W}$ = weight of fluid which passed through nozzle, in pounds. v_j = the velocity of the jet, in feet per second.

Also, from Fig. 16,

$$\text{(12)} \qquad \textit{Kinetic energy of stream leaving block} = \frac{\mathbf{W}v_e^2}{64.4} \quad \text{(ft.-lb.)}$$

Now since, as explained, $v_e = v_j - 2v_b$:

$$\text{(13)} \qquad \textit{Kinetic energy of stream leaving block} = \frac{\mathbf{W}(v_j - 2v_b)^2}{64.4} \qquad \text{(ft.-lb.)}$$

Hence,

(14) *Kinetic energy converted into work* = (*Kinetic energy of jet*) − (*kinetic energy of stream leaving block*)

Or using the symbols instead of words:

(15) *Kinetic energy converted into work* =

$$\frac{\mathbf{W}v_j^2}{64.4} - \frac{\mathbf{W}v_e^2}{64.4} = \frac{\mathbf{W}}{64.4}(v_j^2 - v_e^2) \qquad \text{(ft.-lb.)}$$

13. A Perfect Steam Turbine Would Convert All The Liberated Heat Into Mechanical Work.—Thus, all of the heat energy would first be converted into kinetic energy and then, in turn, into mechanical work. Obviously, then, for a perfect steam turbine:

(16) *Mechanical Work* = *Heat liberated*

Substituting, then, the expression for the heat liberated from For. (5):

(17) *Mechanical Work* = W = $778\mathbf{W}(\mathbf{H}_1 - \mathbf{H}_2)$ (ft.-lb.)

Wherein: W = the work done, in ft. lb. $\mathbf{W}$ = the weight of steam used, in the "perfect" turbine, in pounds. $\mathbf{H}_1$ = the total heat of the steam admitted to the perfect turbine, in British thermal units per pound; this may be found from the steam chart of Fig. 15. $\mathbf{H}_2$ = the total heat of the steam after adiabatic expansion to the pressure at which it is exhausted from the perfect turbine, in British thermal units per pound; this may also be found from Fig. 15 as explained under Sec. 10.

14. The Horsepower And Water Rate Of A Perfect Steam Turbine are dependent on the conditions of the steam admitted to the turbine and on the pressure at which the turbine exhausts steam; the horsepower also depends on the rate at which steam is supplied to the turbine; that is, in a sense, on the capacity of the boiler. The formulas for the horsepower and water rate of a perfect turbine which are derived below, follow:

(18) $$\mathbf{P} = \frac{778\mathbf{W}\,(\mathbf{H}_1 - \mathbf{H}_2)}{60 \times 33{,}000} = \frac{\mathbf{W}(\mathbf{H}_1 - \mathbf{H}_2)}{2{,}545} \qquad \text{(horsepower)}$$

(19) $$\mathbf{W}_H = \frac{2{,}545}{\mathbf{H}_1 - \mathbf{H}_2} \qquad \text{(lb. per hp.-hr.)}$$

Wherein: $\mathbf{P}$ = the power of the perfect turbine, in horsepower. $\mathbf{W}$ = the rate of steam supply, in pounds per hour. $\mathbf{H_1}$ = the total heat of the steam under the conditions at which it is supplied to the perfect turbine, in British thermal units per pound. $\mathbf{H_2}$ = the total heat of the steam after adiabatic expansion to the exhaust pressure, in British thermal units per pound. $\mathbf{W}_H$ = the weight of steam flow required for the perfect turbine in pounds per horsepower per hour; this is the *water rate.*

Derivation.—Using For. (17) to express the work done by $\mathbf{W}$ pounds of steam, there results:

$$W = 778\mathbf{W}(\mathbf{H_1} - \mathbf{H_2}) \quad \text{(ft.-lb.)} \tag{20}$$

But, since in For. (20), $\mathbf{W}$ expresses the weight of steam used in 1 hr., W expresses the work done in 1 hr. Now, since a rate of 1 hp. = 33,000 ft.-lb. per min., also obviously 1 hp. = 60 × 33,000 ft.-lb. per hour. Therefore, to find the number of horsepower in W of For. (20), it is only necessary to divide For. (20) by 60 × 33,000; thus:

$$\mathbf{P} = \frac{728\mathbf{W}(\mathbf{H_1} - \mathbf{H_2})}{60 \times 33{,}000} = \frac{\mathbf{W}(\mathbf{H_1} - \mathbf{H_2})}{2{,}545} \quad \text{(horsepower)} \tag{21}$$

which is the same as For. 18. Now, by transposing For. (21):

$$\mathbf{W} = \frac{2{,}545 \times \mathbf{P}}{\mathbf{H_1} - \mathbf{H_2}} \quad \text{(lb. per hr.)} \tag{22}$$

Dividing For. (22) by $\mathbf{P}$:

$$\frac{\mathbf{W}}{\mathbf{P}} = \frac{2{,}545 \times \mathbf{P}}{(\mathbf{H_1} - \mathbf{H_2})\mathbf{P}} \quad \text{(lb. per hp.-hr.)} \tag{23}$$

But, since $\mathbf{W}/\mathbf{P} = \mathbf{W}_H$, For. (23) simplifies to:

$$\mathbf{W}_H = \frac{2{,}545}{\mathbf{H_1} - \mathbf{H_2}} \quad \text{(lb. per hp.-hr.)} \tag{24}$$

which is the same as For. (19).

Example.—A theoretically perfect steam turbine is supplied with dry saturated steam at 175 lb. per sq. in. gage pressure and exhausts into a condenser where the vacuum is 28.5 in. of mercury column. The turbine is supplied with steam at the rate of 1,000 lb. per hour. What are its horsepower and water rate? Solution.—From the chart of Fig. 15: $\mathbf{H_1}$ = 1,197 B.t.u. per lb. $\mathbf{H_2}$ = 851 B.t.u. per lb. By For. (18): $\mathbf{P} = \mathbf{W}(\mathbf{H_1} - \mathbf{H_2})/2{,}545 = 1{,}000 \times (1{,}197 - 851) \div 2{,}545 = 136.0$ *hp.* By For. (19): $\mathbf{W}_H = 2{,}545/(\mathbf{H_1} - \mathbf{H_2}) = 2{,}545 \div (1{,}197 - 851) = 7.36$ *lb. per hp.-hr.*

Note.—The Theoretical Water Rate Of Any Steam Turbine May Be Read From A Chart, *AB*, Fig. 15. The theoretical water rate of any turbine is, of course, the water rate of a perfect turbine operating

under the same steam conditions. In Fig. 15, values of W_H, as computed by For. (19), have been shown in scale A opposite the values of $H_1 - H_2$ on scale B from which they were calculated. The use of scales A and B of Fig. 15, therefore, replaces the use of For. (19).

15. The Horsepower And Water Rate Of A Commercial Steam Turbine depend in part on the same factors as do those of a perfect steam turbine but, in addition, they depend on the *efficiency* of the turbine in its conversion of heat energy into mechanical work. Energy is "lost" in all steam turbines; that is, some energy is not converted into work. Some of the losses are: (1) *Frictional losses at the surfaces over which the*

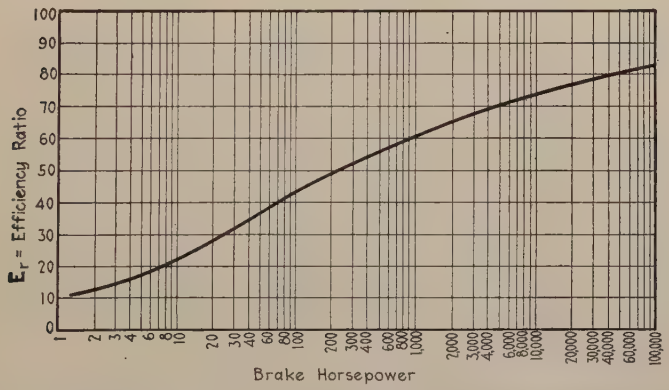


FIG. 20.—Graph showing approximate values of the efficiency ratio, based on brake horsepower, for commercial steam turbines at rated full load.

steam rubs. (2) *Eddy losses*, which are introduced whenever the current of the steam suffers an abrupt change in direction, as when the steam current strikes anything but the desired surface. (3) *Windage losses;* these are occasioned by the motion of the turbine parts within a space which is filled with steam vapor. (4) *Radiation losses;* that is, the heat which is lost as such from the hot surfaces. (5) *Frictional losses in the bearings.* (6) *Exhaust losses*, due to velocity which is not converted into work. (7) *Leakage losses* introduced when steam flows through the turbine, or part of it, without passing through the desired path.

The formulas for the horsepower and water rate of a commercial turbine follow directly from those of Sec. 14 by introducing the efficiency and are:

$$\mathbf{P}_B = \mathbf{E}_r \frac{\mathbf{W}(\mathbf{H}_1 - \mathbf{H}_2)}{2{,}545} \qquad \text{(horsepower)} \tag{25}$$

$$\mathbf{W}_B = \frac{2{,}545}{\mathbf{E}_r(\mathbf{H}_1 - \mathbf{H}_2)} \qquad \text{(lb. per hp. hr.)} \tag{26}$$

Wherein: $\mathbf{P}_B$ = the brake (or delivered) power of the turbine, in horsepower. $\mathbf{E}_r$ = the "efficiency ratio" or efficiency of the commercial turbine as compared with that of a perfect turbine, expressed decimally; approximate values of $\mathbf{E}_r$ are given in Fig. 20. $\mathbf{W}_B$ = the water rate of the commercial turbine, in pounds per brake horsepower-hour. The other symbols having the same meanings as in Sec. 14.

Example.—A 200-hp. commercial steam turbine operates on dry saturated steam at 175 lb. per sq. in. gage and exhausts into a condenser where the vacuum is 28.5 in. of mercury column. What, approximately, is its water rate at full load and at what rate must it be supplied with steam to develop its full load rating? Solution.—As in the example under Sec. 14: $\mathbf{H}_1$ = 1,197 B.t.u. per lb. $\mathbf{H}_2$ = 851 B.t.u. per lb. From Fig. 20, for a 200-hp. turbine, $\mathbf{E}_r$ = 49 per cent = 0.49. Hence, by For. (26): $\mathbf{W}_B = 2{,}545/[\mathbf{E}_r \times (\mathbf{H}_1 - \mathbf{H}_2)] = 2{,}545 \div [0.49 \times (1{,}197 - 851)] = 2{,}545 \div 169.5 = 15.01$ *lb. per b.hp.-hr.* At 200 *hp.* the turbine will require, as is found by combining and simplifying Fors. (25) and (26): $\mathbf{W} = \mathbf{P}_B \times \mathbf{W}_B = 200 \times 15.01 = 3{,}002$ *lb. per hr.*

16. Theoretically, The Work Done By An Impulsive Jet (From A Stationary Nozzle) Is a Maximum If The Velocity Of The Moving Surface Which It Strikes Is One-half That Of The Jet (for the conditions shown in Fig. 16). The proof of this is given below. Actually, the velocity of the moving surface must always be slightly less than one-half that of the jet.

Proof.—It is evident by inspection of Fig. 16 and For. (15) that the kinetic energy converted into work will be a maximum when $v_e^2 = 0$; that is when $v_e = 0$. Hence, since by Sec. 11, $v_e = v_j - 2v_b$, when $v_e = 0$ then $v_j - 2v_b = 0$. Or, then, by transposing: $2v_b = v_j$ or $v_b = v_j/2$. Hence, it is proved that the work done by the jet is a maximum when $v_b = v_j/2$; that is, when the velocity of the moving surface is one-half that of the jet. This result seems logical for (Fig. 16) when $v_b = v_j/2$

then $v_e = 0$ and, then from For. (12), the *kinetic energy of the stream leaving the block* = 0; therefore, the total kinetic energy of the jet has been converted into work.

NOTE.—THE WORK DONE BY A REACTION JET (FROM A MOVING NOZZLE) IS A MAXIMUM WHEN THE VELOCITY OF THE NOZZLE IS EQUAL TO THAT OF THE JET. It is obvious that, in order that the actual kinetic energy of the steam leaving a reaction wheel (Fig. 1) be zero, the nozzle must travel away from the steam jet as fast as the steam passes through the nozzle. Then, since no kinetic energy remains in the steam, it must all have been converted into work.

17. "Staging" In A Steam Turbine Is The Division Into Parts Of The Processes Of Converting Heat Energy Into Mechanical Work. As explained in the previous section, the kinetic energy of a jet can be fully utilized by conversion into work only when the velocity of the moving surface (in an impulse turbine; see Sec. 30) is approximately one-half that of the jet; or, in a reaction turbine (Sec. 31) when the velocity of the nozzles is approximately equal to that of the jet. Furthermore, the velocity of a steam jet is very great (see example under Sec. 10 wherein the theoretical jet velocity is 51 miles per minute). Such high steam velocities would in actual steam turbines necessitate extremely high velocities of moving surfaces or nozzles (rotating parts); in fact, structural difficulties prohibit such velocities except in very small turbines. These high steam velocities can, however, be either prevented or utilized by "staging."

18. There Are Three Classes Of "Staging," velocity, pressure and reaction. Each is defined below; see also the explanations which follow.

1. PRESSURE STAGING (Fig. 21) is that in which the conversion of the available heat energy of the supplied steam into mechanical work is divided into the desired number of steps by causing the steam to expand through two or more impulsive-jet nozzles successively or in series, from each of which the steam is directed against moving surfaces. There will be as many "steps" (pressure stages, Sec. 40) as there are stationary nozzles; in Fig. 21 *II* there are 4 steps.

EXPLANATION.—THE EFFECT OF PRESSURE STAGING may be understood by a study of the hydraulic analogy shown in Fig. 21. Suppose that the level of the water in the reservoir, *R*, is just 156 ft. above the nozzle *A*. Then water will issue from *A* at a velocity of approximately 100 ft. per sec. Hence, the velocity of the blades or buckets against

which *A* directs its stream should, for maximum work, be approximately ½ of 100 or 50 ft. per sec. (Sec. 16). Suppose, however, that the pressure which produces the velocity is divided by the arrangement of Fig. 21 *II*, so that each of the nozzles *B*, *C*, *D* and *E* is supplied from a tank in which the water level is only one-fourth as high above the nozzle as in *I*. Then each nozzle will deliver water at a velocity of approximately 50 ft. per sec., and the velocity of the blades or buckets of each wheel in *II* need only be 25 ft. per sec. Theoretically, arrangements *I* and *II* will develop the same amount of work from a given quantity of water. Practically, arrangements *I* and *II* will give very nearly the same amount of work.

In a pressure-staged steam turbine, the principle described above is exactly duplicated by arrangements (as are shown in Div. 3) which divide the liberation of heat by the steam into a satisfactory number of steps. The kinetic energy produced by each liberation of heat is converted into work before the next liberation of heat.

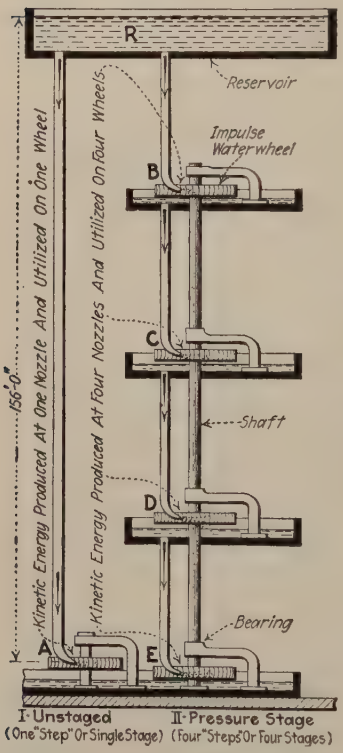


Fig. 21.—A pressure-staged hydraulic turbine. (Analogous to a pressure-staged impulse steam turbine.)

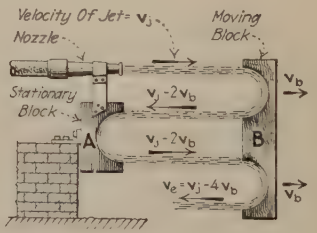


Fig. 22.—Illustrating the principle of velocity staging. (Two velocity "steps" or stages.)

2. Velocity Staging (Fig. 22) is that in which the conversion of the available heat energy, of the supplied steam, into mechanical work is divided into the desired number of steps by using a single impulse-jet nozzle and then, after the jet leaves the first moving surface, redirecting it with guide vanes against additional moving surfaces. There will then be as many "steps" (velocity stages, Sec. 39) as there are moving surfaces over which the steam must pass; in Fig. 22 there are two moving surfaces, hence two steps.

EXPLANATION.—THE EFFECT OF VELOCITY STAGING is illustrated in Fig. 22. If, instead of being used as in the arrangement of Fig. 16, a stream be reversed in direction by a stationary block, *A* (Fig. 22) and thus redirected against a second moving surface on the block, *B*, the jet will again have its velocity reduced by twice the velocity of the moving surface. Thus, in Fig. 22, the velocity of the stream as it finally leaves the moving block, *B*, is $v_e = v_j - 4\,v_b$. Hence, for maximum work, $v_e = 0$ and $v_b = v_j/4$. Thus, if $v_j = 100$, $v_b = 25$. Comparing this with Fig. 16, where (Sec. 16) for maximum work $v_b = v_j/2$, it is obvious that the block in Fig. 22 (which represents buckets on an impulse-turbine rotor) need travel only half as fast as that in Fig. 16, for if in Fig. 16, $v_j = 100$ then, for maximum work, $v_b = 50$.

3. REACTION STAGING (Fig. 40) is that in which the conversion into work of the available heat energy in the supplied steam is divided into the desired number of steps by causing the steam to expand through a successive series of two or more moving reactive-jet nozzles. There will be as many steps as there are reactive-jet nozzles.

QUESTIONS ON DIVISION 1

1. Define a *heat engine*. Is a steam turbine a heat engine?

2. Give a brief history of the development of the steam turbine and draw sketches to illustrate Hero's and Branca's turbines.

3. What is the first step in the conversion of heat energy in a steam turbine? Give an everyday example of the physical change involved in this first step.

4. Describe the second step in the conversion of heat energy in a steam turbine. In this second step does the action of steam differ from that of any other fluid? Why?

5. Cite several common examples of impulsive forces. Draw a sketch to show how an impulsive force may be measured. What primitive steam turbine utilized impulsive forces only?

6. Give several common examples of reactive forces. Draw a sketch to show how a reactive force may be measured. What primitive steam turbine utilized reactive forces only?

7. What sort of force is produced when a fluid stream strikes an object and then leaves it in an opposite direction? Draw a sketch to show how this force may be measured. What kinds of turbines are typical examples of the use of such forces?

8. How is it shown that steam liberates heat when it flows through an opening from a region of high pressure to one of lower pressure? What becomes of this heat?

9. What relation holds between the kinetic energy which steam acquires in flowing through an opening and the heat energy which is liberated?

10. State the formula for the theoretical velocity of a steam jet. Show its derivation.

11. How is the actual velocity of the jet related to the theoretical?

12. Explain the use of the total-heat-entropy diagram for calculating steam velocities.

13. Explain fully, with a sketch, the reduction of velocity of a fluid stream as it passes over a moving surface. What is the relation between the velocity reduction and the velocity of the moving surface?

14. Does a fluid stream gain or lose kinetic energy as it passes over a moving surface? Explain fully.

15. In a perfect steam turbine, what is the relation between work done and heat liberated? State as a formula.

16. What factors determine the horsepower and water rate of a perfect steam turbine? State and show the derivation of the formulas.

17. Explain the use of the chart of Fig. 15 for finding the theoretical water rate of a steam turbine.

18. Name and describe six forms of energy loss in a commercial steam turbine.

19. State the formula for the water rate of a commercial steam turbine.

20. Explain fully the meaning of the *efficiency ratio* of a steam turbine. What, in general, determines the efficiency ratio of a turbine? What values does it have?

21. What should be the velocity of a moving surface to insure that a fluid stream in passing over the surface will do the maximum amount of work on the surface? Explain fully and show the kinetic energy changes.

22. What should be the velocity of the nozzles of a reaction turbine to provide that the steam will do as much work as possible? How is this explained?

23. How great may the velocity of a steam jet be under some circumstances? State why such high steam velocities lead to difficulties in turbines.

24. Explain how high steam velocities may be either prevented or utilized.

25. What is the underlying principle of pressure staging? Of velocity staging?

PROBLEMS ON DIVISION 1

1. How much heat is theoretically liberated from each pound of steam that flows through an opening from a region where the pressure is 175 lb. per sq. in. gage and the steam is superheated by 20° F. to a space at atmospheric pressure?

2. How much heat is liberated (theoretically) when dry saturated at 100 lb. per sq. in. gage flows through a nozzle into a region where the vacuum is 28 in. of mercury column by gage?

3. In Prob. 1 what theoretical velocity does the steam attain? Compute by formula and compare with result obtained from *BC*, Fig. 15.

4. In a perfect turbine, how much mechanical work would be derived from each pound of steam in Prob. 2?

5. If a perfect turbine with steam conditions as given in Prob. 2 consumes 2,000 lb. of steam per hour, what are its horsepower and its water rate? Compare the result with *AB*, Fig. 15.

6. What might be expected as the water rate of a 2,000-hp. commercial steam turbine which operates under the conditions of Prob. 1 and how much steam would it require per hour at full load?

7. At what velocity should a moving surface (similar to Fig. 16) travel to extract the maximum amount of energy from the jet of Probs. 1 and 3?

DIVISION 2

STEAM-TURBINE NOMENCLATURE AND CLASSIFICATION

19. The Terms Which Are Applied To The Various Kinds Of Steam Turbines And To Their Principal Parts will be defined and illustrated in this division. Terms descriptive of turbines and their parts are used with different meanings by various writers and manufacturers. It is therefore important that the reader understand the meanings which will be implied by the terms as used in this book; hence these definitions. Where several terms are popularly used for the same thing, all will be given; the one which is preferred and which will be used in this book will be stated first.

NOTE.—The principal parts of the turbine will first be defined and illustrated in Secs. 20 to 28. Then the various classes and types as regards their construction and the steam conditions for which they are designed will be defined and illustrated in Secs. 29 to 46.

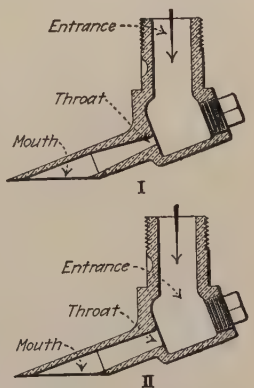


FIG. 23.—*De Laval* divergent nozzles. *I*-Nozzle used in class "C" turbine for high-pressure condensing service. *II*-Nozzle used in class "C" turbine for low-pressure condensing or high-pressure non-condensing service.

20. A Nozzle (Fig. 23) is an opening through which steam is allowed to flow from a region of high pressure to one of lower pressure so as to acquire additional velocity (Sec. 2). The function of a nozzle in an impulse turbine (Sec. 30) is to admit the steam to the active or moving parts of the turbine. In a reaction turbine, the stationary nozzles admit steam to the moving parts which are also of nozzle shape and guide the steam

from them. Nozzles are usually so constructed that the steam flow is restricted by a small opening or *throat* which is the smallest section of the nozzle. The steam is discharged at the *mouth* of the nozzle. Steam always expands in flowing through a turbine nozzle; its expansion is attended with least friction if the nozzle is larger where the steam enters it than at the throat. If the throat of the nozzle is smaller than the mouth, the nozzle is said to be *divergent* (Fig. 23). Nozzles for large pressure-drops are always made divergent. Such divergent nozzles, are sometimes called *Curtis* or *De Laval nozzles.* If the mouth of a nozzle is of the same cross-sectional area as the throat, the nozzle is said to be *convergent* (Fig. 24). Such convergent nozzles are sometimes called *Rateau nozzles.* The nozzles of a turbine are frequently formed by the openings between the blades as in Fig. 24.

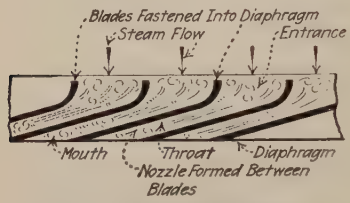


FIG. 24.—Convergent nozzles in diaphragm.

NOTE.—DIVERGENT NOZZLES ARE SOMETIMES CALLED "EXPANDING" NOZZLES; and, similarly, convergent nozzles, *non-expanding.* Since expansion occurs in nozzles of both types, these terms are not strictly correct and should be avoided.

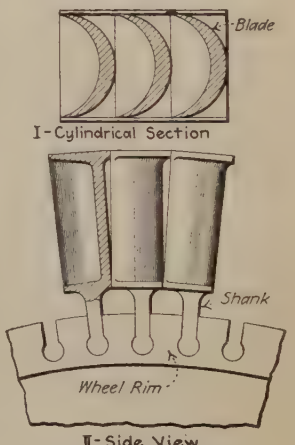


FIG. 25.—Moving blades used in class "C" *De Laval* turbine. These blades are formed by the drop-forging process and the bulb shanks are accurately machined to fit the corresponding recesses in the wheel rim.

21. Blades Or Vanes (Fig. 25) are curved metallic parts, the function of which is to deflect or change the direction of a current or jet of steam. Blades are sometimes called *buckets;* but buckets are, more properly, the deflecting surfaces of a bucket-wheel or tangential-flow turbine (Sec. 43). Blades

may be either *moving blades* on which the work of the steam is done, or *fixed or stationary blades* (Fig. 26) which reverse the direction of the steam jet so that more work may be abstracted from it. Stationary blades are sometimes called *guide vanes*

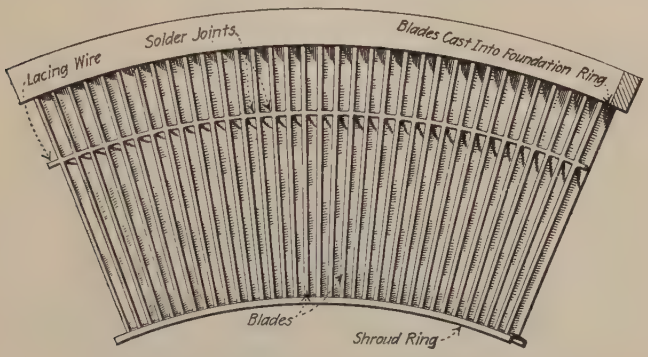


Fig. 26.—Fixed blades of *Allis-Chalmers* Parsons turbine.

or *guide blades*. The openings between the blades frequently constitute nozzles as in Fig. 24.

22. The Rotor Or Runner (Fig. 27) of a turbine is the main moving part which carries the blades or buckets. It consists

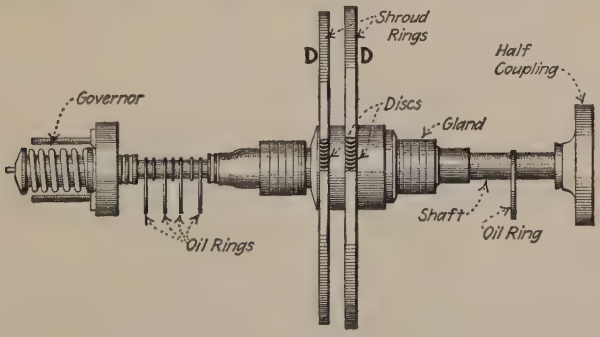


Fig. 27.—Complete rotor with two discs.

mainly of a *spindle or shaft* which is supported by the bearings and which carries one or more *discs, D,* (Fig. 27) *drums* (Fig. 45) *or wheels W,* (Fig. 31) according to the type of turbine. The blades or buckets are carried on the discs, drums or wheels.

23. A Casing, Case Or Housing (Fig. 28) of a turbine is a covering—usually a horizontally split cast-iron shell—which confines the steam and also acts as a frame for the support of the rotor bearings. Casings are frequently provided with

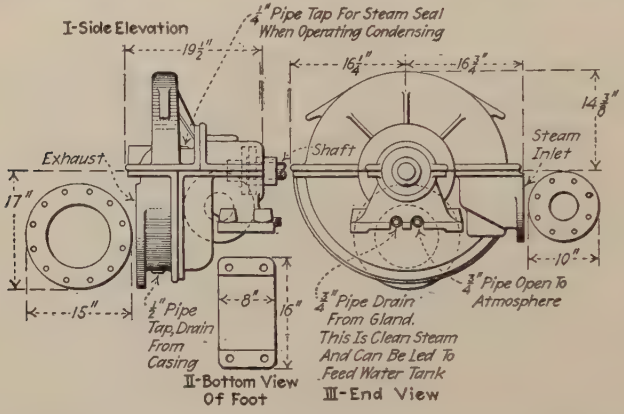


Fig. 28.—Outline dimensions of *Steam Motors Company* turbine casing, frame No. 5, Type B with one bearing pedestal.

relief valves (Fig. 29) to prevent rupture due to excessive pressure. The part of the casing immediately surrounding the moving blades, together with the fixed blades and nozzles

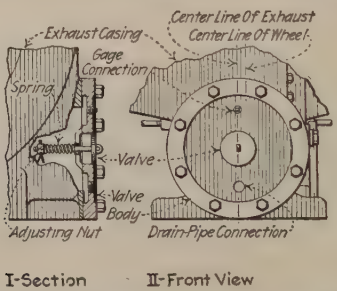


Fig. 29.—Relief valve of Type-6 *Sturtevant* turbine. This is located on the exhaust casing directly opposite the exhaust-pipe opening.

which it carries is sometimes called the *stator* (Fig. 30).

24. A Cylinder (Fig. 30 shows a half cylinder) is a cylindrical part of a casing in which a number of the stationary blades of the turbine are secured. The term cylinder is most frequently used in connection with reaction-type turbines (Sec. 31).

25. A Barrel (*B*, Fig. 30 shows the stationary nozzles of one barrel) is a group of rotor and stator blades which are mounted in rings or drum sections of the same diameter, which are the same height, and are so arranged as to act successively on the steam current. There may be a number of barrels in one turbine cylinder. The term barrel is most frequently used in connection with reaction-type turbines.

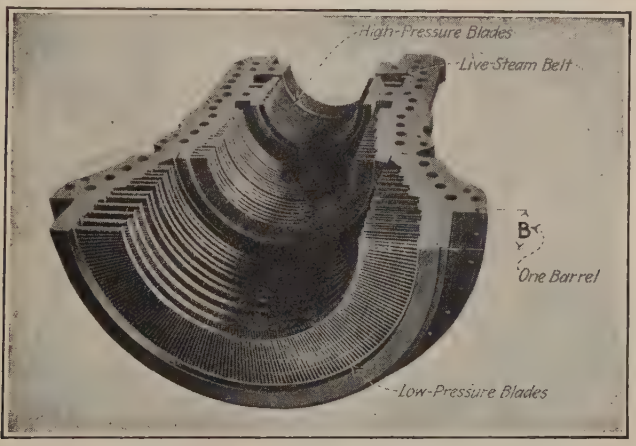


FIG. 30.—Half cylinder—or half stator—of a multi-stage reaction turbine. (Parsons type, *Allis-Chalmers Mfg. Co.*) This turbine has 38 stages.

26. A Gland (*G*, Fig. 31) is a device for preventing the leakage of steam or air between the stationary parts of a turbine and the shaft or the drums which form balance pistons. See Div. 5 for further definitions and examples.

27. A Governor (*C* and *B*, Fig. 31) sometimes called the *speed governor* is a device for maintaining the speed of a turbine practically constant at all loads; see also Div. 6. Governors are either *direct governors* if the centrifugal force of the weights which they employ is the only force used in operating the governing valve; or *indirect or relay governors* if some other force is used to operate the governing valve. An *overspeed governor*, *emergency governor* or *safety stop* (*E* and *V*, Fig. 31) is a device which operates to stop the turbine when its speed exceeds a certain pre-determined value for which the overspeed governor has been set; but which is inoperative as long as this value is not exceeded. (See Div. 6.)

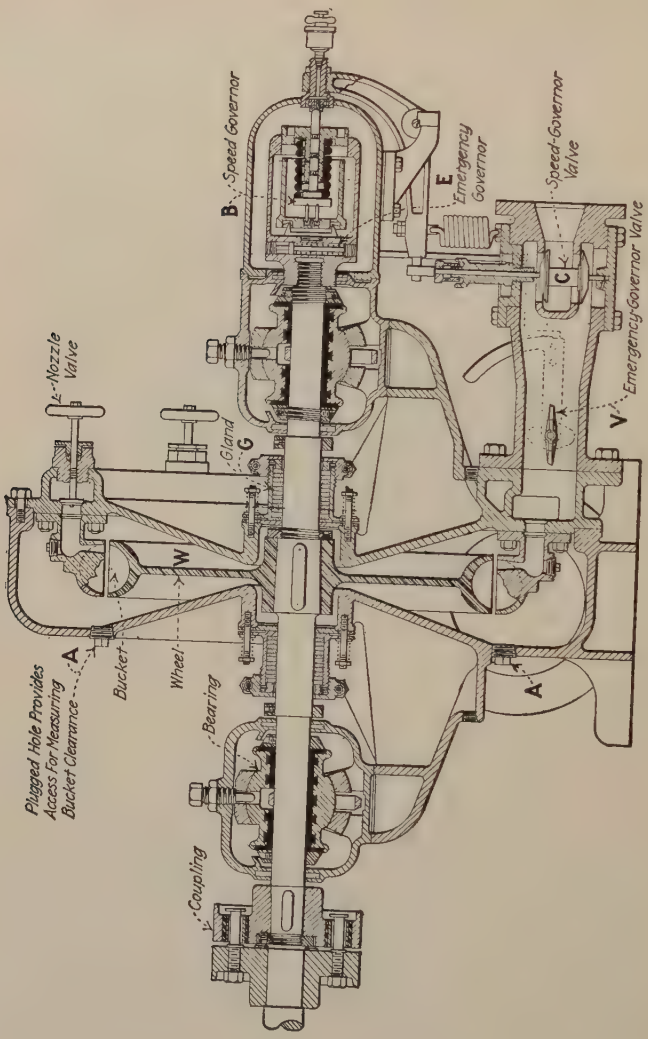


Fig. 31.—A *Sturtevant* bucket-wheel turbine.

28. A Turbine Reduction Gear (Fig. 32) is a mechanism for delivering power to a driven machine at a rotative speed considerably less than the speed of the turbine shaft. (See Div. 7.)

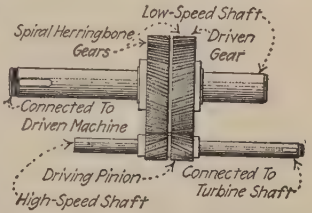


FIG. 32.—A turbine reduction gear.

29. Table Showing The Five Principal Ways, A To E, In Which Steam Turbines May Be Classified and the various classes into which they are thus divided. The terms which describe these classes will be defined in subsequent sections. Study the note on page 30.

Classification	Class with respect to		Class No.	Class	Illustration
A	Blading or nozzle arrangement		1	Impulse.	Fig. 33
			2	Reaction.	Fig. 35
			3	Impulse-and-reaction.	Fig. 83
B	Service or steam conditions		4	High-pressure, non-condensing.	Fig. 57
			5	High-pressure, condensing.	Fig. 69
			6	Low-pressure, condensing.	Fig. 79
			7	Mixed-pressure.	Fig. 38
			8	Bleeder or extraction.	Fig. 39
C	Kinds of stages	Single	9	Single pressure and velocity stage (axial flow).	Fig. 41
			10	Impulse-ree-ntry (axial flow).	Fig. 53
			11	Impulse tangential (bucket-wheel).	Fig. 42
		Velocity stages only	12	Single-pressure, several velocity stages.	Fig. 33
		Reactions tages only	13	Many reaction stages.	Fig. 30
		Pressure stages only	14	Several pressure (impulse) stages one velocity stage in each (multicellular) (Rateau).	Fig. 67
		Pressure and velocity stages	15	Several pressure stages with several velocity stages on one or more (Composite) (Curtis and Rateau).	Fig. 73
		Reaction and velocity stages	16	Impulse-and-reaction turbine.	Fig. 83
D	Direction of flow		17	Axial flow.	Fig. 41
			18	Tangential flow.	Fig. 42
			19	Radial flow.	Fig. 43
E	Division of flow		20	Single flow.	Fig. 44
			21	Double flow.	Fig. 45
			22	Single-and-double flow.	Fig. 46

Note.—Every Turbine Belongs To One Of The Classes Of Each Classification, *A* to *E* inclusive, shown in the preceding table. For example, considering classification, *A*, every turbine is either an impulse, a reaction or an impulse-and-reaction turbine. Also in classification, *D*, it is either axial, radial, or tangential flow. The figure numbers given do not, necessarily, indicate the only illustrations in this volume of each of the various classes (see also Table 70). In fact, some of the classes include a large number of kinds and makes—class No. 17 probably includes over half of the steam turbines now in use. The names of the various manufacturers who make turbines of these various classes and the sizes in which they are made are given in Table 70.

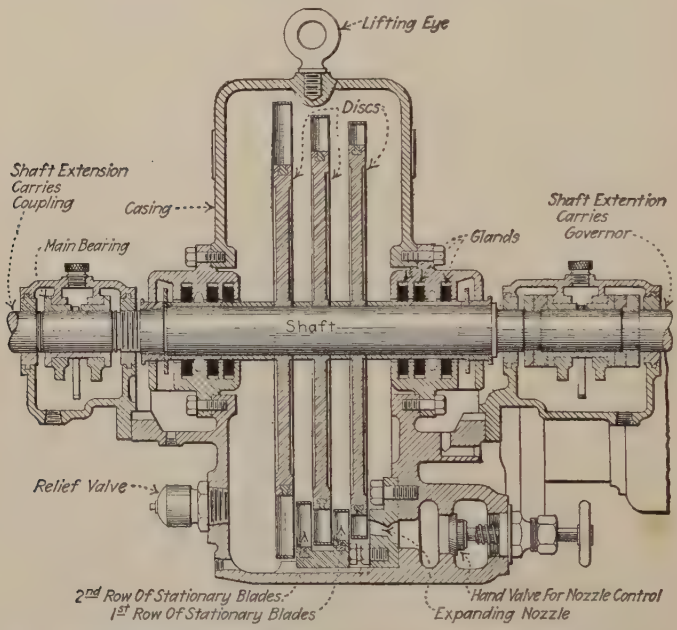


Fig. 33.—Impulse turbine which has three velocity stages in one pressure stage. (*Moore* turbine.)

30. An Impulse Turbine, also called a *velocity turbine* or an *equal-pressure turbine* (Figs. 33 and 34), is one which depends almost wholly for its operation on the "impulsive force" of a steam jet or jets which impinge upon the buckets of the turbine rotor. See Sec. 5 for the definition of an "impulsive force." Thus, an impulse turbine is so designed that the expansion of the steam which passes through it—and makes it

do work—occurs almost entirely in its stationary nozzles or in its fixed blades; practically no expansion of the steam occurs in its moving blades. For an impulse turbine, the designer intends that the steam jet from the stationary nozzles or blades *shall* impinge on the rotor vanes and thus cause the rotor to revolve by virtue of the "push" thus produced. The usual impulse turbine probably operates about 99.5 per cent. by "impulse" and 0.5 per cent. by reaction.

Note.—The Pressure Of The Steam Entering The Moving Blades Of An Impulse Turbine Is Almost Exactly The Same As That Of

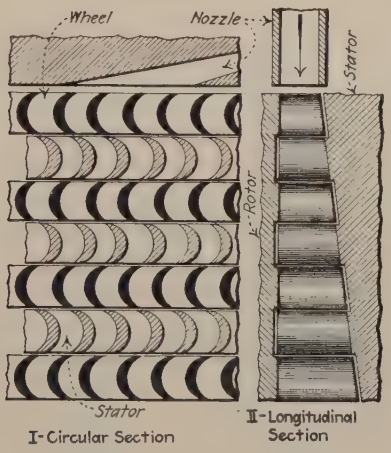


Fig. 34.—Impulse-turbine blading.

The Steam Leaving Them.—This follows since there is no expansion of the steam in the moving blades; see Fig. 40, *I* and *II*. In the nozzles or fixed blades, the steam velocity increases as the steam pressure falls while in the moving blades the velocity of the steam is expended in turning the rotor.

Note.—The Important Characteristics Of Impulse-type Turbines are: Few stages, expansion occurs only in stationary nozzles, large drop in pressure per stage, best efficiency is obtained when blade velocity s approximately one half the initial velocity of the steam (Sec. 16).

31. A Reaction Turbine, also called an *unequal-pressure turbine* (Figs. 35 and 44), is one which depends principally on the "reactive force" of the steam jets as they leave the

turbine's revolving blades at greater velocities than those at which they approached the blades. See Sec. 7 for the definition of a "reactive force." Thus, a reaction turbine is so designed that about half of the expansion of the steam which passes through it and causes it to do work occurs in the moving blades—and about half in the stationary guide vanes. For reaction turbines, the designer endeavors to so design the guide vanes and moving blades that the steam will flow into the blades without striking them. This he does by endeavoring to insure that the circumferential speed of the moving blades will be the same as the velocity of the steam stream which enters them. But the moving blades are so designed that the steam leaves them at a higher velocity than that at which it entered them; thus the rotation of the rotor is produced by reaction. The usual reaction turbine probably operates about 90 per cent. by "reaction" and 10 per cent. by "impulse." Hero's turbine Fig. 1 was a purely reaction turbine.

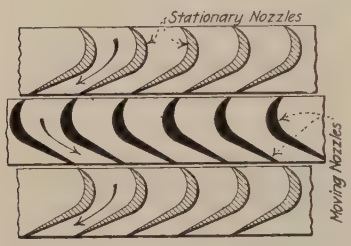


FIG. 35.—Reaction-turbine blading. The space between the adjacent blades, on the rotor and on the stator, form slightly-divergent nozzles.

NOTE.—THE PRESSURE OF THE STEAM ENTERING THE MOVING BLADES OF A REACTION TURBINE IS HIGHER THAN THAT OF THE STEAM LEAVING THEM. This follows because expansion occurs in the moving blades; see Fig. 40-*III*. Some of the heat energy of the steam is changed to mechanical work (Sec. 2) in passing through the moving blades.

NOTE.—THE IMPORTANT CHARACTERISTICS OF REACTION-TYPE TURBINES are: Many stages, expansion occurs in moving as well as in stationary nozzles, small pressure-drop in each stage, best efficiency is obtained when blade velocity is nearly equal to the highest steam velocity (Sec. 16).

32. The Distinguishing Difference Between Impulse And Reaction Turbines is, therefore, that: In the impulse turbine there is no appreciable expansion of steam in the moving blades; in reaction turbines there is considerable expansion of the steam in the moving blades. Furthermore, it follows

that: In impulse turbines there is practically no difference between the pressure of the steam which enters the moving blades and that of the steam which leaves them; in a reaction turbine there is a difference between these entering and leaving pressures.

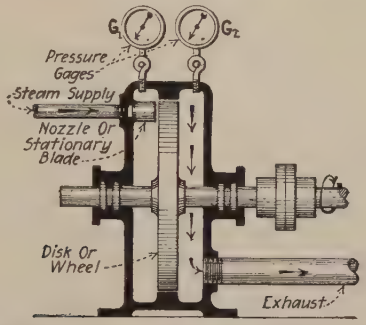


FIG. 36.—On impulse turbines, G_1 reads the same as G_2. On reaction turbines G_1 reads higher.

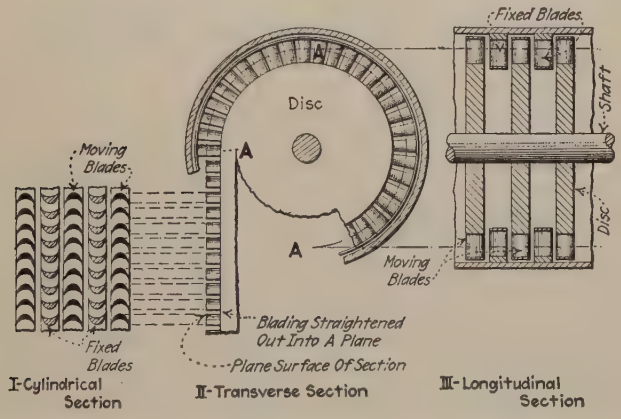


FIG. 37.—Showing relation of a "cylindrical section" to the actual blading of a turbine.

NOTE.—TO DETERMINE WITH PRESSURE GAGES WHETHER A TURBINE IS OF THE IMPULSE OR THE REACTION TYPE, take steam-pressure-gage readings G_1 and G_2, as in Fig. 36. If there is no difference between the readings, the turbine is of the impulse or equal-pressure type, because in this type there is no pressure drop in the moving blades. If G_1 is greater than G_2, the turbine is of the reaction or unequal-pressure type, in which type there is a steam-pressure drop in the moving blades.

NOTE.—THE DISTINGUISHING DIFFERENCE BETWEEN IMPULSE BLADING AND REACTION BLADING is that the cross-sectional shape of impulse

blades (Fig. 34-*I*) is such that the exterior curved surfaces of adjacent blades in a row, lie almost parallel to one another whereas the curved surfaces of reaction blades are such that the opening between adjacent blades is smaller on the steam outlet than on the inlet ride, thus forming a nozzle.

NOTE.—A CYLINDRICAL SECTION (Fig. 37) also called a *circumferential or circular section* is employed in Figs. 34 and 35, and in many other pictures in this book, in illustrating steam flow in turbine blading. A cylindrical section is a section which is taken along a cylindrical surface through the turbine blading; it shows as a circle, *AAA* (Fig. 37) in the end view or transverse section. This surface *AAA* is considered to be unrolled or unbent as shown in *II* and then appears, when looking at it from the side, as a "cylindrical section." The blades in a cylindrical section all show their true cross sections and spacing, whereas any plane section through the blades would cut some of them obliquely and show the sections distorted.

33. An Impulse-and-reaction Turbine (Fig. 83) is one which has some of its blading designed and arranged as in an impulse turbine and some as in a reaction turbine. See Secs. 30 and 31. Some of the largest turbines now in use are of this type. The impulse blading is used for the first stages as will be explained later.

34. High-pressure Non-condensing And Condensing Turbines are turbines which are designed to operate on steam at 100 to 350 lb. per sq. in. pressure and exhaust at atmospheric pressure or into a condenser respectively. The chief structural difference between the two is the much larger exhaust steam spaces of the condensing turbine which are necessary to provide for the large volume of steam at the low pressure of the condenser. Non-condensing turbines which are designed to operate at a back pressure considerably above atmospheric are called *back-pressure turbines.*

35. A Low-pressure Or Exhaust-steam Turbine is one which is designed to operate on low-pressure steam—say 0 to 10 lb. per sq. in. gage. A low-pressure turbine is always a condensing turbine and usually operates on the exhaust from a high-pressure turbine or from a reciprocating engine. The low-pressure turbine is characterized by the large steam spaces near the admission end which are necessary for the large volume which the steam occupies at the low pressure. See Sec. 192.

36. Mixed-pressure Turbines (Fig. 38, also called *mixed-flow turbines*) are turbines to which steam is admitted at two or more pressures. They usually operate on a combination of live steam from the boiler and additional exhaust steam from some other equipment; this exhaust steam being admitted to an intermediate steam belt before the low-pressure stages. Thus, steam from both sources flows through the low-pressure stages. (See Div. 9.)

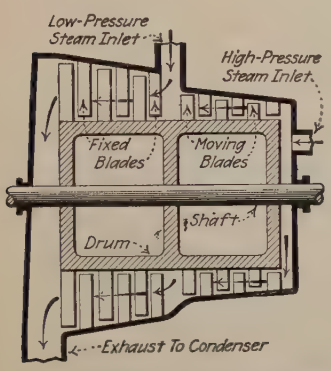


FIG. 38.—Diagram of a mixed-pressure turbine.

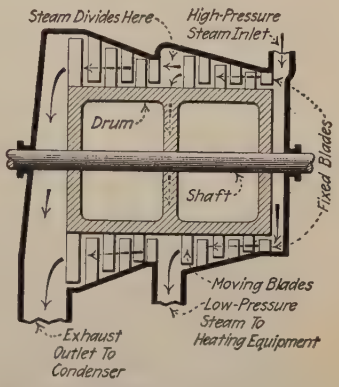


FIG. 39.—Diagram of a bleeder turbine.

37. A Bleeder Turbine Or Extraction Turbine (Fig. 39) is one from which steam is extracted at an intermediate stage and led away to be used for some other purpose, usually for heating. The usual arrangement is to extract enough steam at about atmospheric pressure for feed-water or building heating and to allow the rest to flow through the low-pressure stages of the bleeder turbine and thence to the condenser. Obviously, more steam passes through the high-pressure stages of a bleeder turbine than through its low-pressure stages. (See Div. 9.)

38. A Stage, as defined in general terms, is: A period constituting a development or one of several well defined successive periods in a development. A *steam-turbine stage* may be defined as a section which comprises, or one of a number of well defined sections which comprise, the steam path through a turbine. This general definition, however, is indeterminate because it does not fix the limits of the section which comprises

the stage. Different kinds of stages are spoken of in connection with turbines but their meanings are not definitely established nor recognized among manufacturers and writers. The following definitions are proposed because they are

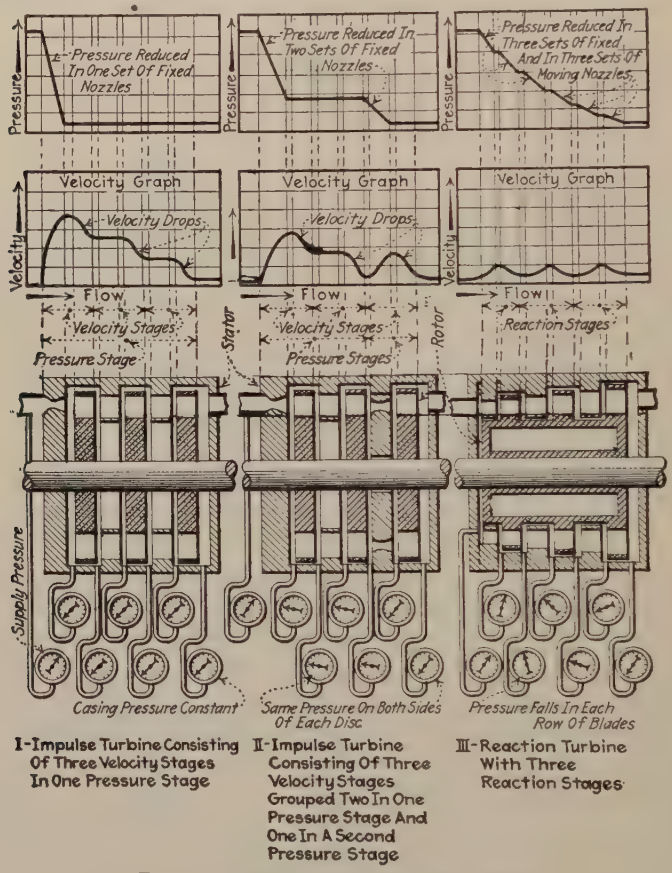

Pressure
Pressure Reduced In One Set Of Fixed Nozzles
Pressure Reduced In Two Sets Of Fixed Nozzles
Pressure Reduced In Three Sets Of Fixed And In Three Sets Of Moving Nozzles
Velocity
Velocity Graph
Velocity Drops
Flow
Velocity Stages
Pressure Stage
Stator
Pressure Stages
Rotor
Reaction Stages
Supply Pressure
Casing Pressure Constant
Same Pressure On Both Sides Of Each Disc
Pressure Falls In Each Row Of Blades
I-Impulse Turbine Consisting Of Three Velocity Stages In One Pressure Stage
II-Impulse Turbine Consisting Of Three Velocity Stages Grouped Two In One Pressure Stage And One In A Second Pressure Stage
III-Reaction Turbine With Three Reaction Stages

Fig. 40.—Illustrating different kinds of stages.

consistent with the most general use of the terms and are sufficiently distinct not to be confusing. Different terms are used for impulse and reaction turbines because the processes are different in the two.

39. A Velocity Stage (Fig. 40–*I*) is that portion of the steam path in a turbine, wherein work is done by the impulsive force

of the steam—see Sec. 5 for definition of "impulsive force"—which consists of one row of stationary nozzles (or one set of stationary guide vanes) and the moving blades of the one runner which immediately follows the row of nozzles or vanes and on which the steam from the nozzles impinges. A velocity stage may begin with one row of either nozzles or guide vanes and always includes only *one* set of moving blades.

40. A Pressure Stage (Fig. 40–*II*) is that portion of the steam path, in a turbine, wherein work is done by the impulsive force of the steam, which comprises one or more velocity stages through which the steam passes consecutively, its first velocity stage having nozzles and the other velocity stages being all which follow up to the next set of nozzles. A pressure stage always begins with a set of nozzles but may contain in addition many rows of impulse stationary guide vanes and corresponding rows of moving blades.

41. A Reaction Stage (Fig. 40–*III*) is that portion of the steam path, in a turbine, wherein work is done by the reactive force of the steam, Sec. 7, which is composed of a set or row of stationary nozzles and that row of moving blades upon which these nozzles direct the steam. The steam, in passing through a reaction stage, suffers a reduction of pressure in both the stationary and the moving blades. Reaction stages are frequently called pressure stages but it is believed to be better to reserve the latter name for the use given in Sec. 40. A half-cylinder of a reaction turbine with 38 reaction stages is shown in Fig. 30.

42. Various Terms Which Are Used To Designate The Staging Of Impulse Turbines and their significance are as follows:

Single-stage Turbines (Fig. 31) are those impulse turbines which are composed of but one pressure stage which contains but one velocity stage.

Velocity-staged Turbines (sometimes called velocity-stage turbines), Fig. 33, are those impulse turbines which are composed of but one pressure stage which contains two or more velocity stages.

Pressure-staged Turbines (sometimes called pressure-stage turbines) are those impulse turbines which are composed of two or more pressure stages each of which contains but one velocity stage.

VELOCITY-AND-PRESSURE-STAGED TURBINES are those impulse turbines which are composed of two or more pressure stages each of which contains two or more velocity stages.

COMPOSITE-STAGED TURBINES are those the stagings of which are formed as a combination of some of the above stagings, so arranged that the steam passes through them in succession or in series: see Sec. 63.

For a further discussion of staging see Div. 3.

43. An Axial-flow Turbine (Fig. 41) is one in which the steam flows in a direction approximately parallel to the rotor

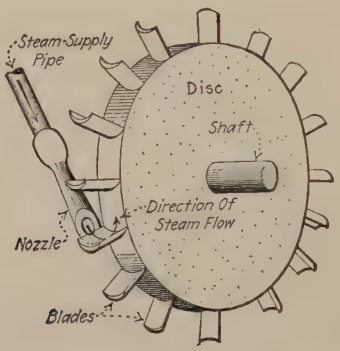


FIG. 41.—Elementary diagram of an axial-flow, single-stage turbine.

axis. Nearly all large turbines and many small ones are of this type. A *tangential-flow turbine*, also called a *bucket-wheel*

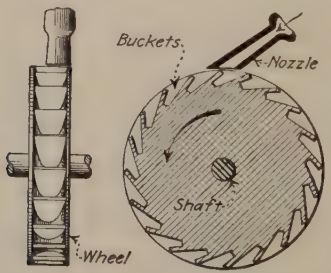


FIG. 42.—Elementary diagram of a tangential-flow turbine.

turbine (Figs. 31 and 42) is one in which the flow of steam is approximately tangent to the rim of the wheel. Many small turbines are of this type. A *radial-flow turbine* (Fig. 43) is one in which the flow of steam is radially inward toward or

outward from the shaft. Radial-flow turbines have never been regularly manufactured in America but have been

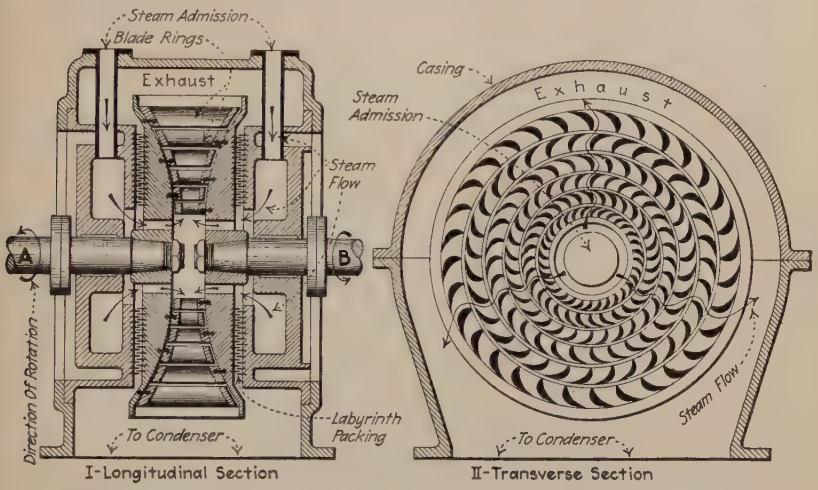


Fig. 43.—Diagram showing action of steam in *Ljungstrom* radial-flow reaction turbine. Shafts *A* and *B* are forced to rotate in opposite directions; each drives its own generator.

built in Europe by a Swedish engineer; one is being built in the United States.

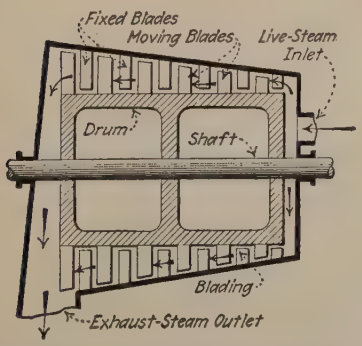


Fig. 44.—Elementary single-flow reaction turbine.

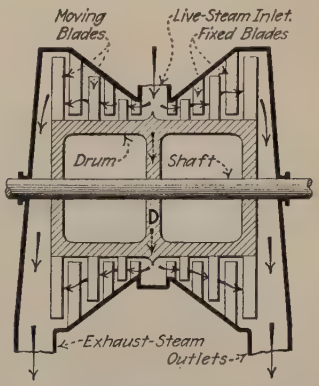


Fig. 45.—Elementary double-flow reaction turbine.

44. A Single-flow Turbine (Fig. 44) is one in which nearly all of the steam which drives the turbine flows together

through the blades in the same general direction parallel to the rotor axis.

45. A Double-flow Turbine (Fig. 45) is one in which the main steam current is divided and the parts flow parallel to the rotor axis in opposite directions.

46. A Single-and-double-flow Turbine or *semi-double-flow turbine* (Fig. 46) is one in which the steam flows through part

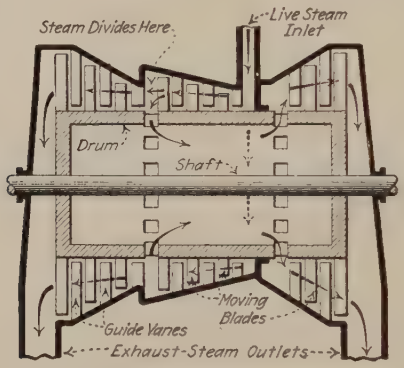


Fig. 46.—Elementary single-and-double flow turbine.

of the blades as a single current, then divides into two currents which flow in opposite directions.

QUESTIONS ON DIVISION 2

1. What are two general classes of nozzles? Name three parts of a nozzle.

2. In what two ways are blades used in steam turbines? What is the function of the guide vanes in an impulse turbine?

3. What is a rotor? What are its principal parts?

4. What are some of the functions of a casing? How is rupture of casings by pressure prevented?

5. Define the terms, *barrel* and *cylinder* as used in steam turbine nomenclature.

6. What is the purpose of a gland?

7. What are speed governors? Relay governors? What other kinds of steam-turbine governors are there?

8. Why is a turbine reduction gear used?

9. Name four ways in which steam turbines may be classified. Give at least three classes under each classification.

10. What is an impulse turbine? Is it actuated entirely by impulse?

11. What is a reaction turbine? What other force does it employ?

12. What is the chief difference between the action of the steam in impulse blading and in reaction blading?

13. What is the direction of flow in a bucket-wheel turbine? What is meant by *axial flow*?

14. What are *pressure stages? Velocity stages?* What are the differences between the two?

15. What is the chief difference in design between a condensing and a non-condensing high-pressure turbine?

16. What are the usual sources of steam for a low-pressure turbine?

17. What is the approximate pressure range for the steam supply for a high-pressure turbine? For a low-pressure turbine?

18. What is the difference between a bleeder turbine and a mixed-pressure turbine? To what other equipment may the steam outlets of each be connected.

19. What is a double-flow turbine? A semi-double-flow turbine? Explain with sketches.

DIVISION 3

STEAM-TURBINE TYPES AND CONSTRUCTION

47. Table Showing Classification Of Steam Turbines By General Construction.—This classification will be followed in the development of this division. Note that this classification combines in one arrangement all of the five classifications, *A* to *E*, Table 29.

Blading Or Nozzle Arrangement (A, Table 29)	Staging Or Cylinder Arrangement (C, Table 29)	Type	Direction Or Divison Of Flow (D And E, Table 29)		Usual Steam Conditions (B, Table 29)
Impulse	Single Stage	De Laval	Single Entry	Axial Flow	High-Pressure, Condensing And Non-Condensing
			Re-Entry	Axial Flow Tangential Flow	High-Pressure, Condensing Or Non-Condensing; Or Low-Pressure, Condensing
	Velocity-Staged	Curtis	Single Entry	Axial Flow	High-Pressure, Condensing Or Non-Condensing; And Low-Pressure Condensing
	Pressure-Staged	Rateau			
	Velocity-And-Pressure-Staged	Curtis			
	Composite Staged	Curtis And Rateau			
Reaction	Single Cylinder	Parsons	Single-Flow Double-Flow Single-And-Double Flow		High-Or Low-Pressure, Condensing
	Compounded	Parsons	Cross	H.P. Element, Single Flow; L.P. Element, Double Flow, Or Single-And-Double Flow	High-Pressure Condensing
			Tandem		
Impulse-And-Reaction	Single Cylinder	Curtis And Parsons	Single Flow Double Flow Single-And Double Flow		High-Or Low-Pressure, Condensing
	Compounded	Curtis And Parsons	Cross	H.P. Element, Single Flow; L.P. Element, Double Flow, Or Single-And-Double Flow	High-Pressure Condensing
			Tandem		

48. The Three Fundamental Types Of Steam Turbines are, see Table 47: (1) *Impulse*, Sec. 30, and Fig. 50. (2) *Reaction*, Sec. 31 and Fig. 78. (3) *Impulse-and-reaction*, Sec. 33 and Fig. 83. The principal features which are embodied in the construction of steam turbines of each of these types are described hereinafter in this division.

NOTE.—STEAM TURBINES ARE MANUFACTURED IN BOTH THE HORIZONTAL AND VERTICAL TYPES. In a "horizontal" turbine, the shaft is, horizontal. In a "vertical" turbine, the shaft is vertical. However, vertical steam turbines, though formerly widely used, are, except in small sizes for driving sump pumps and similar services, becoming obsolete. Step-bearing troubles rendered vertical turbines unreliable. Therefore, only horizontal turbines will be discussed in this division. The general construction, except bearings, of both types is similar.

49. The Four Principal Types Of Impulse Steam Turbines are (Table 47): (1) *Single stage*, Sec. 42 and Fig. 50. (2)

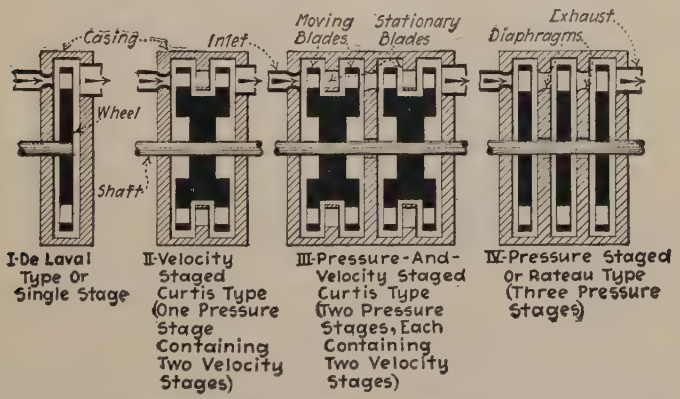


FIG. 47.—Illustrating De Laval, Curtis and Rateau types of steam turbines.

Velocity-staged, Sec. 42 and Fig. 63. (3) *Pressure-staged*, Sec. 42 and Fig. 67. (4) *Velocity-and-pressure-staged*, Sec. 42 and Fig. 70. As shown in Table 47 and in the following sections, certain of these types may be still further subdivided. Also, two types of impulse-turbine staging—usually (2) and (3)—may be combined in one turbine. A turbine which is made up of such a combination of staging is (Sec. 42) called *composite-staged*.

Note.—Impulse Steam Turbines May Also Be Classified According To The Name Of The Man Who Obtained The Original Patents (Table 47) as: (1) *De Laval.* (2) *Curtis.* (3) *Rateau.* A single-stage, axial-flow turbine (Fig. 47-*I*) is usually considered to be of the De Laval type. Those turbines which belong either to the velocity-staged type (Fig. 47-*II*) or to the velocity- and pressure-staged type (Fig. 47-*III*)

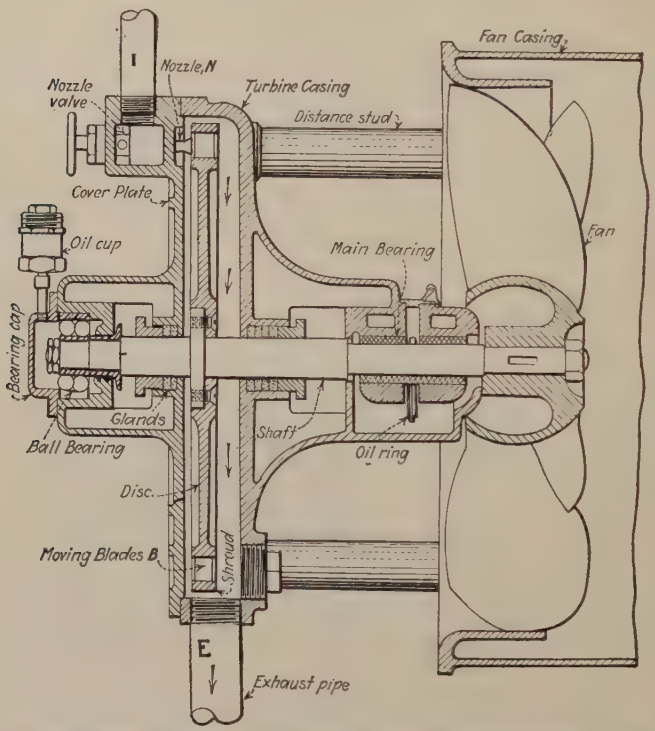


Fig. 48.—Single-stage impulse, single-entry turbine direct-connected to a blower, manufactured by the *Power Turbo-Blower Co.*

are generally called the Curtis type. A pressure-staged turbine (Fig. 47-*IV*) is generally considered to be of the Rateau type. From a consideration of *I* and *IV*, it will be noted that, in reality, the Rateau type merely consists of a number of turbines of the De Laval type arranged in series. It should not be inferred that all of the turbines which are manufactured by the *De Laval Steam Turbine Co.* are of the De Laval type. In fact, most of the large-capacity turbines which are manufactured by this company (Table 70) closely resemble the velocity-staged (Curtis) type or the Rateau type.

50. The Two Principal Types Of Single-stage Impulse Steam Turbines (Table 47) are: (1) *The single-entry type*, Fig. 48, wherein the steam jet strikes the moving blades only once. (2) *The re-entry type*, Figs. 54, 57, and 58, wherein the direction of flow of the steam jet is reversed and it is made to strike the same set of moving blades or buckets two or more times. Different manufacturers' single-stage turbines of each of these types are briefly described in the following sections.

51. The Single-stage Single-entry Impulse Steam Turbine (Figs. 48 and 49) is the simplest type of turbine. Because of their inherently high speeds, mechanical difficulties render impracticable the manufacture of single-stage single-entry turbines in capacities greater than about 600 hp. If a single-stage single-entry turbine is run at the proper speed, it is the most efficient of any turbine within its capacity limits—up to about 600 hp. However, this proper speed is so high, that for most services, reduction gears (Div. 7) will be required. Consequently, it is frequently desirable to run a single-stage single-entry turbine at a speed which is much lower than the speed at which it would have the maximum efficiency. This is because that, by running the turbine at a lower speed, the reduction gear may, for these small capacities, sometimes be economically eliminated. Turbines of this type are generally designed to operate at steam pressures from about 100 to 250 lb. per sq. in., with exhaust pressures ranging from about a vacuum of 28 in. of mercury up to 35 lb. per sq. in. gage. Their usual operating speed is some speed between about 2,000 and 5,000 r.p.m. However, some small single-stage single-entry turbines have been designed to operate at about 30,000 r.p.m.

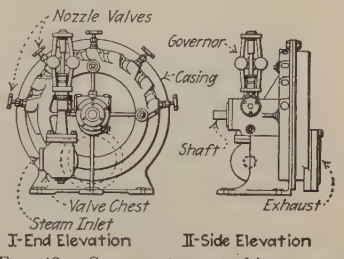


FIG. 49.—*Coppus* steam turbine, type TC.

52. The Usual Construction Of Single-stage Impulse Turbines Of The Single-entry Type (Table 47) is indicated in Figs. 48, 50 and 51 which show turbines manufactured,

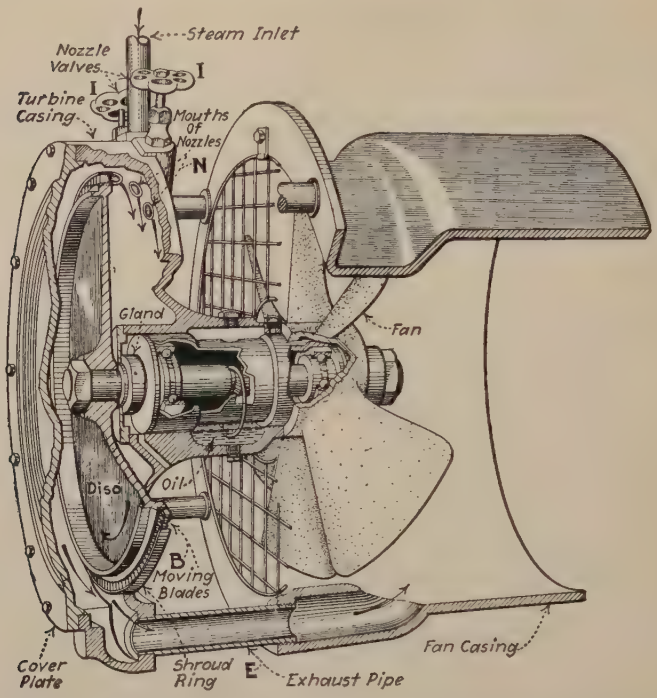


FIG. 50.—Single-stage, single-entry *Coppus* impulse turbo-blower, Type C. (The exhaust *E*, may be so arranged that all, none, or only a part of the exhaust steam will be delivered into the blower.)

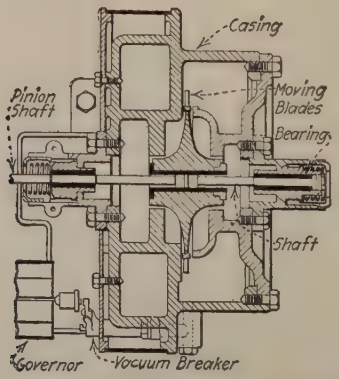


FIG. 51.—Single-entry, single-stage steam turbine. (*De Laval Steam Turbine Co.*)

respectively by the *Power Turbo-Blower Co.*, the *Coppus Engineering & Equipment Co.*, and the *De Laval Steam Turbine Co;* the method of converting the heat energy of the steam into mechanical energy is the same in all of these three turbines. The path of the steam through the turbines is, as indicated by the arrows, from the inlet, *I*, through the nozzles, *N*, through the moving blades, *B*, and out through the exhaust, *E*. As the steam passes through the diverging nozzles, *N*, it expands (Div. 1). This expansion results in a considerable drop in pressure and an increase in velocity of the steam. The pressure drop is practically equal to the difference between the steam pressure at the inlet and that at the exhaust. Practically all of the velocity energy which the steam thus acquires is converted into mechanical work as the steam jet impinges on the moving blades. The steam passes through the moving blades only once.

53. Single-stage Impulse Steam Turbines Of The Re-entry Type (Table 47 and Figs. 54 and 57) are but slightly more complex in construction than those (Sec. 51) of the single-entry type. Because of the fact that the steam strikes their moving blades two or more times (see Sec. 18), turbines of the re-entry type can be operated with but a slightly lower efficiency at a much lower speed than can those of the single-entry type. Turbines of the re-entry type are, in general, used for larger capacities for about the same classes of service as are those of the single-entry type. Single-stage turbines of the re-entry type are made in capacities of from about 1 to 1,000 hp. They are designed to operate at steam pressures from about 75 to 250 lb. per sq. in., and at exhaust pressures ranging from a high vacuum up to about 35 lb. per sq. in. gage. The usual operating speed of turbines of this type is some speed between about 3,000 and 5,000 r.p.m.

54. There Are Two Types Of Single-stage Re-entry Impulse Turbines (Table 47): (1) *Axial flow*, Sec. 43 and Fig. 54. (2) *Tangential flow*, Sec. 43 and Fig. 57. The principle of energy conversion in each type is essentially the same as that of the single-entry turbine (Sec. 51). However, in the re-entry types only a part of the velocity energy of the steam is given up to the rotating wheel the first time it strikes the moving

blades. After the steam has impinged once on the moving blades or buckets it passes through a reversing chamber, which

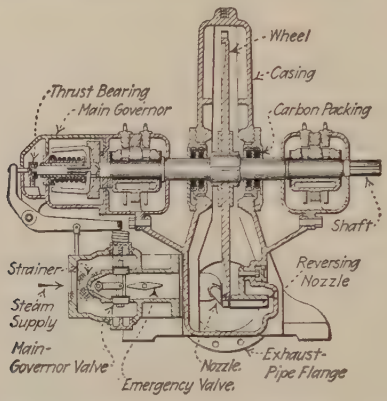


Fig. 52.—*Midwest Engine Company* single-stage, axial-flow, re-entry turbine, **longitudinal section.** See Fig. 54 for detail of reversing nozzle of this turbine.

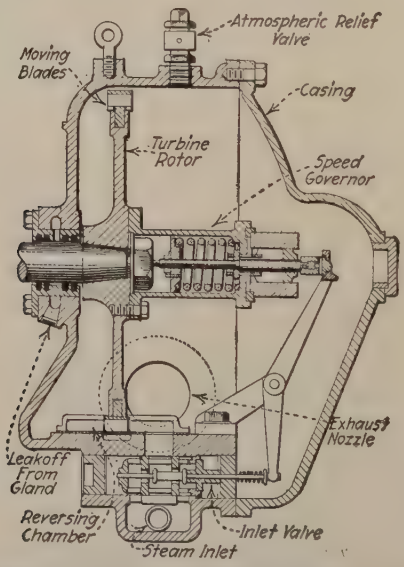


Fig. 53.—Cross section of a 10-kw., single-stage, axial-flow re-entry turbine. (*Westinghouse Electric & Mfg. Co.*)

changes the direction of flow, whereby the steam is made to strike the moving blades a second time. In the axial-flow

type (Fig. 54) the direction of steam flow is usually changed only once; consequently, in this type, the steam-jet strikes the moving blades only twice. In the tangential-flow type,

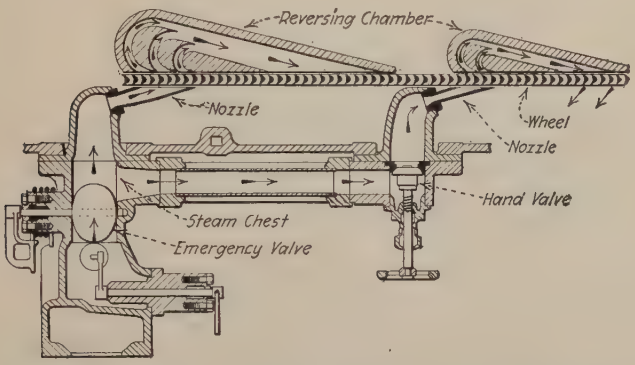


FIG. 54.—Cylindrical section showing arrangement of nozzles and reversing chamber of an impulse, single-stage, axial-flow re-entry turbine. (*Midwest Engine Co.*)

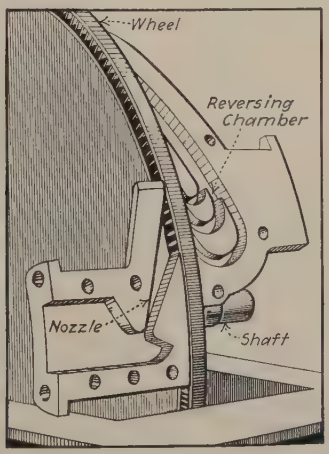


FIG. 55.—Showing nozzle and reversing chamber of an axial-flow, single-stage, re-entry steam turbine. (*Westinghouse Electric & Mfg. Co.*)

the steam jet generally undergoes two or more reversals (Fig. 58), thus striking the moving blades three or more times. Each type is briefly described in the following sections.

55. The Usual Constructional Arrangement Of Impulse Single-stage Re-entry Turbines Of The Axial-flow Type

(Table 47) is indicated in Figs. 52 and 53. The device which is employed to reverse the direction of the steam flow is called the *reversing chamber*. The path of the steam through the nozzles, the moving blades, the reversing chamber and back through

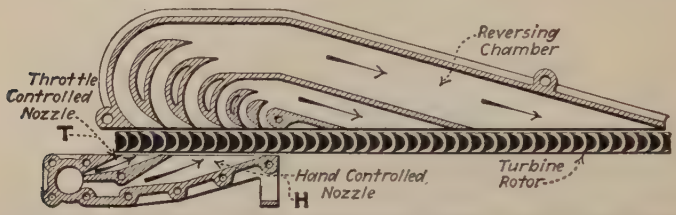


Fig. 56.—Cylindrical section showing arrangement of nozzles which is sometimes used on axial-flow single-stage re-entry turbines. (*Westinghouse Electric & Mfg. Co.*)

the moving blades the second time is indicated by tne arrows in Figs. 54 and 55. Turbines of this type as manufactured by the *Midwest Engine Co.* (Fig. 52) are usually provided with three or more nozzles, two or more of which may be controlled

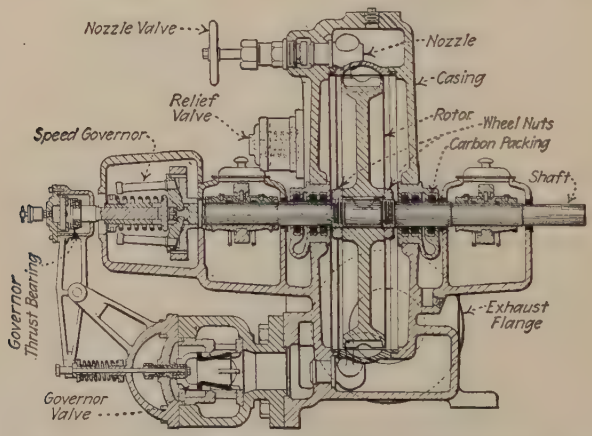


Fig. 57.—*Terry* Type-Z2 turbine, longitudinal section. (Axial adjustment of the wheel is made by the wheel nuts.)

by a hand-operated valve as shown in Fig. 54. The *Westinghouse* turbines of this type usually have only one nozzle (Fig. 55). However, for certain services, some of the *Westinghouse* turbines of this type are provided with two nozzles as shown in Fig. 56. One of the nozzles, *H*, may be

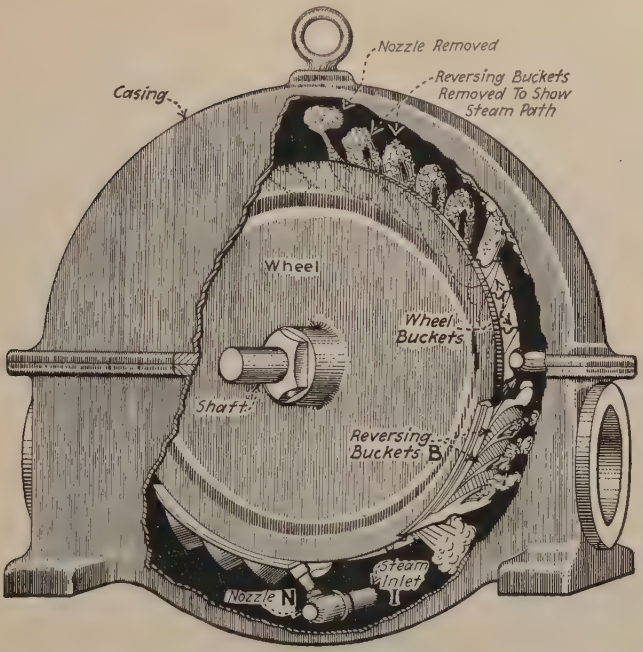


FIG. 58.—Showing path of steam jet in a tangential-flow, single-stage, re-entry turbine; part of the nozzle and reversing bucket is broken away to better show the steam path. See Fig. 59 for the nozzle of a similar turbine. (*Terry Steam Turbine Co.*)

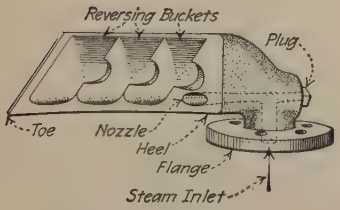


FIG. 59.—Nozzle and three reversing buckets of *Sturtevant* turbine, made from one solid bronze casting.

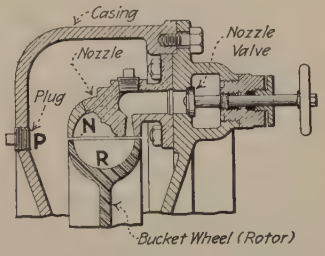


FIG. 60.—Nozzle valve of Type-6 *Sturtevant* turbine. To inspect for proper longitudinal alignment of rotor and nozzle, remove plug *P*. The alignment is correct when the edge of rotor, *R*, is flush with the edge of nozzle *N*.

controlled by a hand valve. The other nozzle, *T*, is controlled by the constant-speed governor.

56. The Usual Construction Of Impulse Single-stage Re-entry Turbines Of The Tangential-flow Type (Table 47)—as manufactured by the *B. F. Sturtevant Co.* and by the *Terry Steam Turbine Co.*—is shown, respectively, in Figs. 31 and 57. The principle of operation, as explained below, is the same in both turbines. About the only difference between the two turbines is in the details of construction.

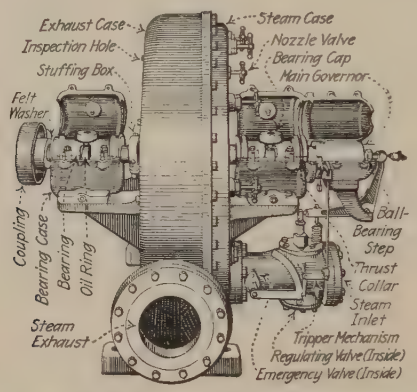


Fig. 61.—External view of Type-6 *Sturtevant* turbine.

Explanation.—The Flow Of The Steam Jet In A Single-stage Impulse Turbine Of The Re-entry, Tangential-flow Type is shown in Fig. 58. A nozzle and a reversing chamber which contains three reversing buckets is shown in Fig. 59. The steam, after passing through the expanding nozzle (*N*, Fig. 58) strikes the side of one of the semi-circular-shaped wheel or rotor buckets. This wheel bucket changes the direction of the steam-flow through 180 deg. The steam jet then strikes the first reversing bucket, *B*, of the stationary reversing chamber. This stationary reversing bucket again changes the direction of the steam flow through 180 deg. so that the steam jet strikes another wheel bucket. This reversal is repeated until practically all of the velocity energy of the steam is converted into mechanical work of turning the wheel, whereupon the steam passes out of the buckets into the casing and then through the exhaust. A cross-section of a nozzle valve for, and an external view of a Type-6, *Sturtevant* turbine are shown, respectively, in Figs. 60 and 61.

57. Impulse Turbine Of The Velocity-staged Type (Table 47 and Figs. 33 and 64) inherently have a lower rotative

speed than do those of the single-stage single-entry type. This is because the velocity-staged turbines employ two or three sets of moving blades (Fig. 62) with a set of stationary blades or guide vanes between each successive pair of moving blades. The steam is, in the nozzles (*N*, Fig. 63), expanded from the initial pressure (Fig. 63-*IV*) and temperature down to the exhaust pressure (Fig. 63-*IV*) and temperature. About one-half of the velocity energy (Fig. 63-*III*) which is thus acquired is, in a velocity-staged turbine having two rows of moving blades, converted into mechanical work in the first row of moving blades. After the steam has passed through this first row of moving blades, the direction of flow is reversed by the stationary blades so that the steam jet strikes the second row of moving blades (Fig. 63-*II*).

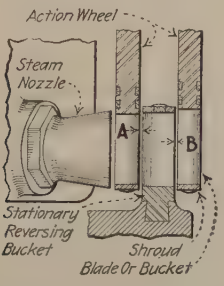
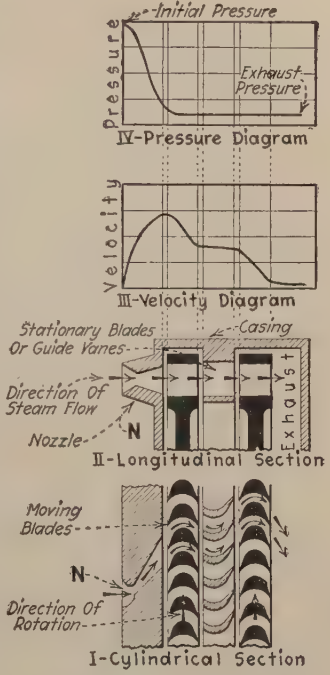


FIG. 62.—Steam nozzle, reversing bucket and action wheel of *Terry* turbine. "On action wheels the *side clearance* is the important factor. Clearances *A* and *B* should be kept approximately equal and neither should be less than $\frac{1}{16}$ in."



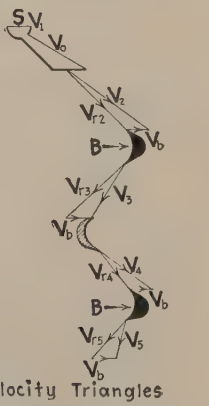


FIG. 63.—Illustrating action of the steam in a velocity-staged turbine which has two velocity stages.

Practically all of the remaining velocity energy is converted into mechanical work in this second row of moving blades. Consequently a velocity-staged turbine which has two rows of moving blades will, when operating under the same conditions, run at about one-half the speed of a single-stage turbine. See also Sec. 18.

Note.—Velocity-staged Turbines Are Not, Inherently, Quite As Efficient As Are Single-stage Turbines. Nevertheless, for some services, it is frequently desirable to use the velocity-staged turbines. This is because that by their use, reduction gears may sometimes be dispensed with. For driving a given low-speed unit, a single-stage turbine with a reduction gear might or might not, depending on conditions, provide a higher efficiency than would a direct-connected velocity-staged turbine. Velocity-staged turbines, however, generally are simpler, operate more quietly, and are lower in first cost than single-stage turbines with reduction gears. Consequently velocity-staged turbines find their widest application in driving relatively low-speed auxiliaries of small capacity where low first cost is of greater importance than is the over-all operating efficiency.

58. In The Usual Constructions Of Velocity-staged Impulse Turbines (Figs. 33, 64 and 65) there is one set of expanding nozzles with either two rows of moving blades and one row of stationary blades (Figs. 64 and 65), or three rows of moving blades and two rows of stationary blades (Fig. 33). More than three rows of moving blades are seldom used in velocity-staged impulse turbines. One method which is employed in securing the blades in turbines of this type is shown in Fig. 66. For mechanical reasons, velocity-staged impulse turbines are only made in sizes up to about 1,000 hp. If made in capacities much larger than 1,000 hp., the wheel diameters and blade lengths become so great that the centrifugal stresses thereby introduced are excessive. Their chief application is for driving power-plant auxiliaries

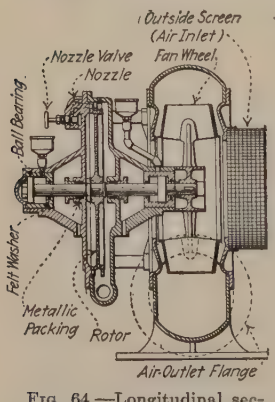


Fig. 64.—Longitudinal section through *Carling* impulse, velocity-staged type, turbine-driven volume fan.

such as centrifugal pumps for circulating cooling water or boiler feeding, blowers, small generators and the like. As manufactured, their speed ratings and steam service condi-

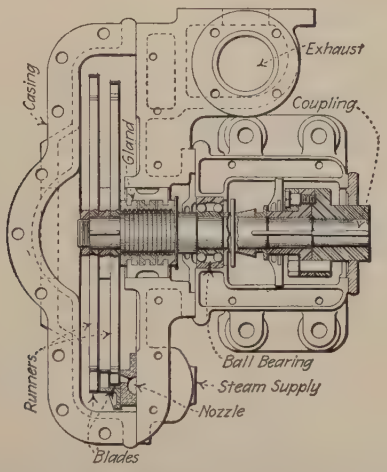


Fig. 65.—*Steam Motors Company* turbine, top view with cover removed and certain parts shown in section. (Steam Motors Company, Springfield, Mass. See Fig. 113 for gland details and Fig. 135 for governor.)

tions are about the same as those given in Sec. 51 for single-stage impulse turbines.

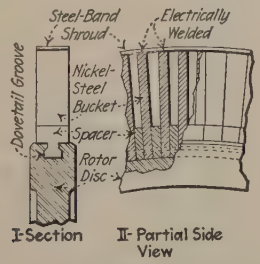


Fig. 66.—Method of attaching buckets to wheel in *Moore* steam turbine.

59. Impulse Turbines Of The Pressure-staged Type (Table 47 and Fig. 67) consist essentially of several single-stage turbines which are contained in one casing and which are connected in series. In the pressure-staged turbines

(Fig. 67) each row of moving blades is separated from the next row of moving blades by a *diaphragm*. This diaphragm con-

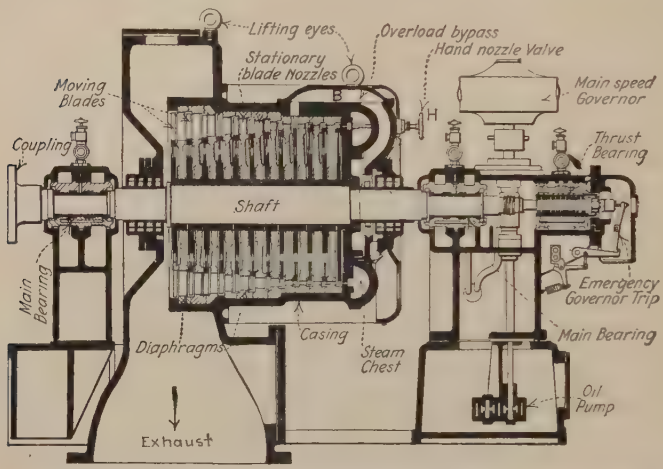


FIG. 67.—Axial section showing general arrangement of a pressure-staged turbine which has 12 pressure stages. (*De Laval Steam Turbine Co.*)

tains stationary blades which are (Sec. 20) of nozzle form. The steam, as it enters the turbine through the first set of nozzles (Fig. 68), is expanded. The velocity which the steam thus acquires is utilized in doing work on the first row of moving blades just as was explained in Sec. 52 for the single-stage turbine. After the steam leaves this first row of moving blades, it passes through the nozzle-shaped stationary blades in the first diaphragm. In passing through these stationary blades a second expansion of the steam, with a consequent velocity increase, occurs. This velocity energy is converted into mechanical work in the second row of moving blades in precisely the same manner as was

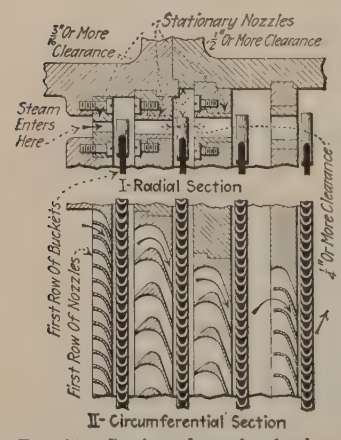


FIG. 68.—Section of nozzles, buckets and wheels of *Ridgway* turbine.

explained for the first row. The action of the steam throughout the succeeding pressure stages is identical to that in either of the first two pressure stages described above.

NOTE.—THE PURPOSE OF PRESSURE STAGING is to provide a method whereby the mechanical difficulties which are encountered in attempting to make a single-stage turbine of large capacity may be surmounted. The velocity of the steam as it issues from a nozzle is a function of the pressure drop (Div. 1). That is, if a large pressure drop occurs, a large velocity increase will result, and if only a small pressure drop occurs, a correspondingly small velocity increase will result. Therefore, by dividing the total pressure drop—inlet pressure minus the exhaust pressure—into a number of small pressure drops, the velocity with which the steam strikes any row of moving blades will be much smaller than if all of the pressure drop was produced in one set of nozzles. Consequently, in a pressure-staged turbine, the velocity and the diameter of the rotor can be decreased and the capacity of the turbine increased over that of the single- or velocity-staged turbine and yet a comparatively high efficiency can be maintained.

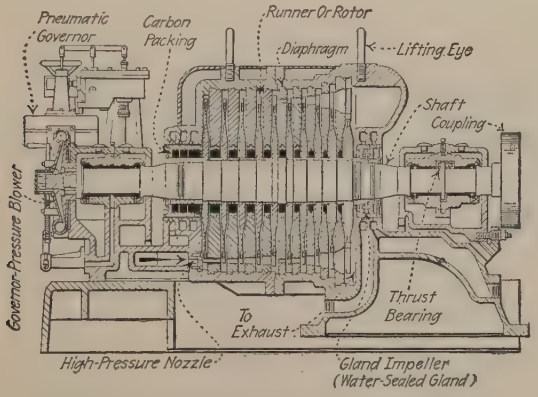


FIG. 69.—Section through *Ridgway* high-pressure turbine.

60. The General Constructional Arrangement Of Impulse Turbines Of The Pressure-staged Type is indicated in Figs. 67 and 69. Although the principle of operation is the same for both of these turbines, the constructional details differ. As indicated in Fig. 68, the clearance between the moving and stationary parts may be comparatively large. In all pressure-staged impulse turbines, some means must be employed to minimize the leakage of steam through the clearance between

the diaphragms and the shaft. This is usually accomplished by a labyrinth passageway or by carbon glands (Div. 5). To take care of overloads, the turbine may be provided with extra nozzles which may be either hand- (*H*, Fig. 67) or governor-controlled, or they may be provided with a bypass (*B*, Fig. 67) so that high-pressure steam may be admitted directly to one of the intermediate stages. Turbines of this type are usually designed for capacities of from about 500 to 5,000 kw., to operate with either high- or low-pressure steam, condensing or non-condensing, at some speed between about 3,000 and 5,000 r.p.m.

61. Impulse Turbines Of The Velocity-and-pressure-staged Type (Table 47, and Fig. 70) consist, essentially, of a

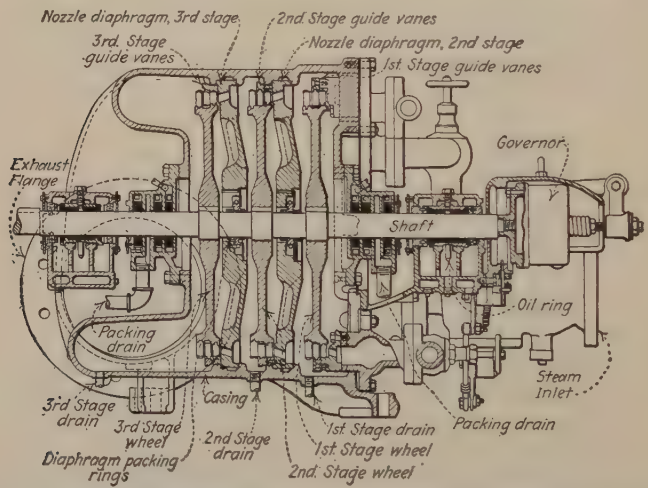


FIG. 70.—Sectional view showing assembly of a velocity-and-pressure-staged impulse turbine which has three pressure stages with two velocity stages in each pressure stage. (*General Electric Co.*)

number of velocity-staged turbines which are contained within the same casing, and which are connected in series. The total steam-pressure drop between the inlet and exhaust is divided into a number of smaller drops as in the pressure-staged turbine (Sec. 59). Then the action of the steam in each pressure stage is the same as that which was described (Sec. 57) for the velocity-staged turbine. The purpose of

velocity-and-pressure staging is about the same as that of pressure staging (Sec. 59). Turbines of this type are also sometimes called the *Curtis* type (Sec. 49).

62. The General Construction Of Impulse Turbines Of The Velocity-and-pressure-staged Type is illustrated in Fig. 70. Turbines of this type are made in capacities of from 10 to 400 hp. and are adapted to operate at pressures from 60 to 250 lb. per sq. in., condensing or non-condensing, at some speed between about 1,200 and 5,000 r.p.m. The non-condensing

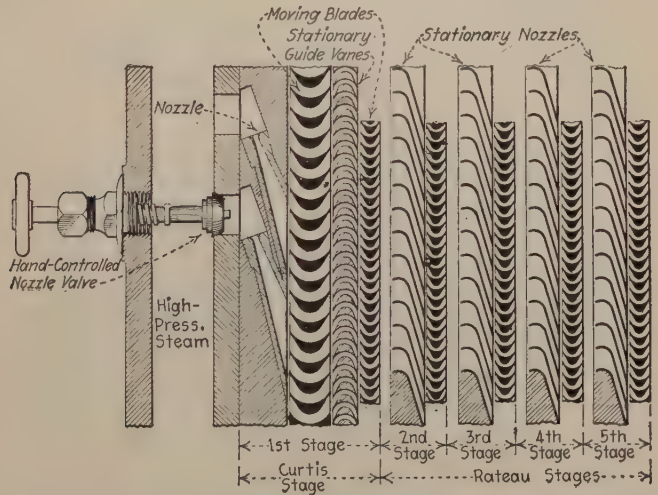


Fig. 71.—Cylindrical section through nozzles and blades of a composite-staged steam turbine. Five stages are shown. (*Moore Steam Turbine Corp.*)

units of this type have two pressure stages. The condensing turbines have two, three or four pressure stages, depending upon the capacity and upon the operating conditions. Each pressure stage has two rows of moving blades and one row of stationary guide vanes. Diaphragms separate the pressure stages from each other. These diaphragms are provided with nozzles, just as are the pressure-staged turbines. Each diaphragm is provided with a metal labyrinth packing to minimize steam leakage along the shaft. Those turbines of this type which have three or four pressure stages may be arranged for either mixed pressure or extraction service (Div. 4).

63. Impulse Turbines Of The Composite-staged Type (Table 47 and Fig. 71) usually consist of a number of pressure stages. The first pressure stage (Fig. 71) usually contains two velocity stages. This first stage is followed by the required number of pressure stages, each of which contains one velocity stage (for exception see Fig. 76). The first stage is sometimes called a Curtis stage, and those which follow are sometimes called Rateau stages. Therefore, a turbine of the

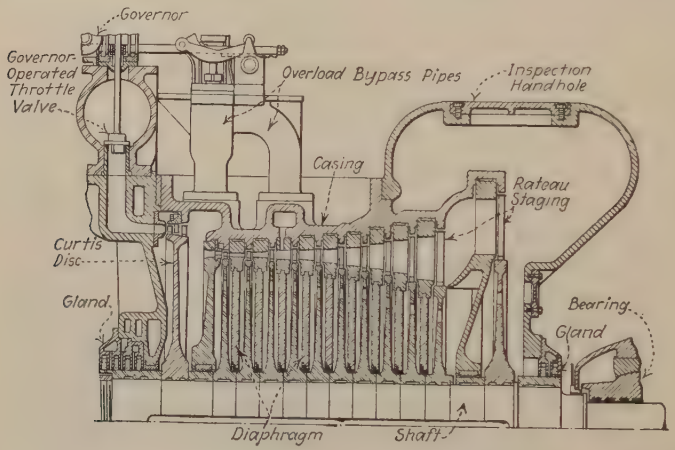


FIG. 72.—Partial longitudinal section of a high-pressure composite-staged impulse turbine which has twelve pressure stages. (*General Electric Co.*)

composite-staged type is, in reality, a velocity-staged turbine (Sec. 57) which has in series with it a pressure-staged turbine (Sec. 59). The action of the steam through such a turbine may be understood from a study of Secs. 57 and 59.

NOTE.—THE REASONS FOR THE USE OF COMPOSITE STAGING in impulse turbines are that, for the larger capacities—above about 1,000 kw.—they are more efficient and less expensive to construct than turbines of any of the types which are described in the preceding sections. This is because the two velocity stages, which are in such turbines always placed in the high-pressure end, will efficiently cover an expansion range equal to several pressure stages. Thus, by employing them, the size and consequently the cost of the turbine may be reduced. Also, by placing the two velocity stages in this first pressure stage, the pressure of the steam therein may be considerably reduced over that which would be required if the velocity staging were replaced by equivalent pressure staging.

This decreases the windage loss and the leakage of steam, thereby increasing the efficiency.

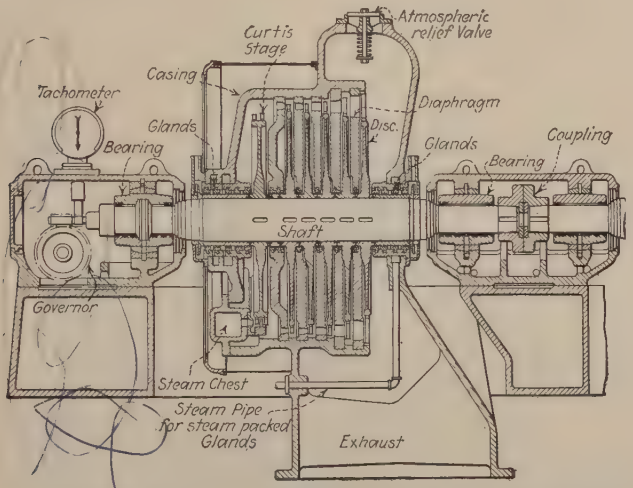


FIG. 73.—Longitudinal section of an impulse turbine of the composite-staged type having one Curtis and five Rateau stages. (*Ingersoll-Rand Co.*)

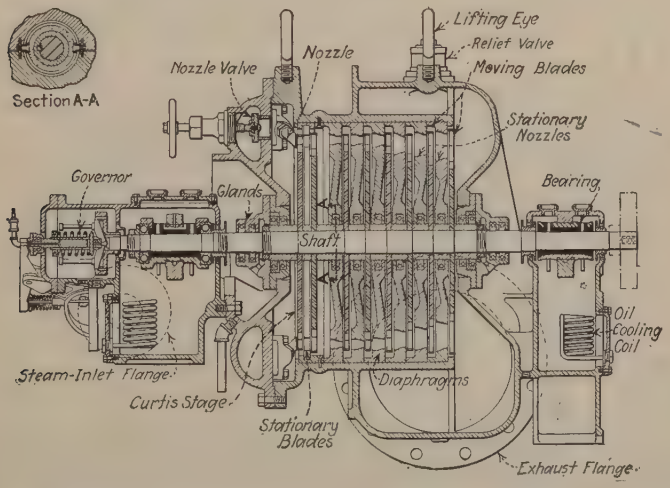


FIG. 74.—Longitudinal section of a composite-staged impulse turbine. (*Terry Steam Turbine Co.*)

64. Various Methods Of Construction Of Composite-staged Impulse Turbines are illustrated in Figs. 72, 73, 74, 75 and

76. The same principle is employed in all of these turbines. However, the different manufacturers follow different mechanical designs. Practically all impulse turbines between

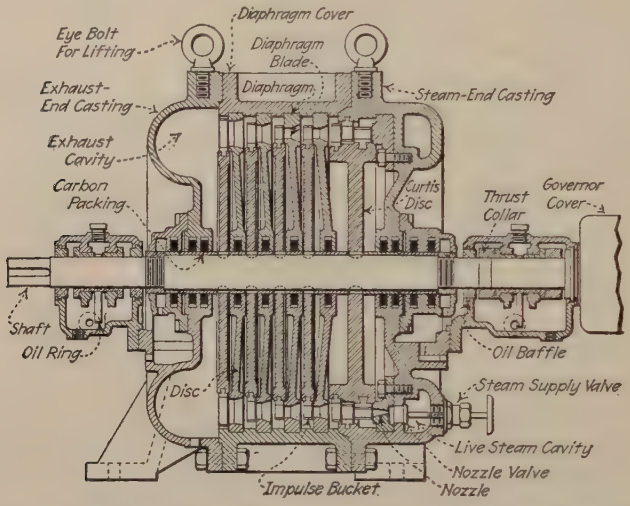


FIG. 75.—Sectional view of *Moore* steam turbine. (Instruction Card No. 1, Moore Steam Turbine Corporation, Wellsville, New York.)

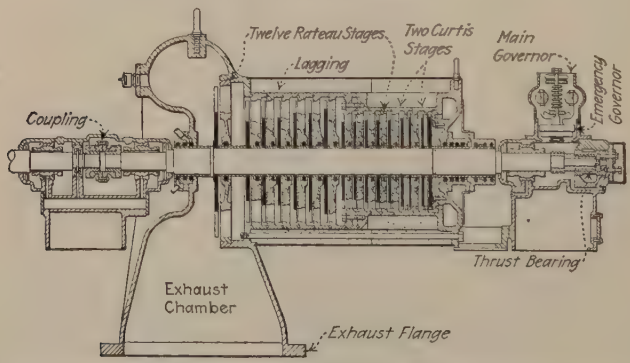


FIG. 76.—Longitudinal section through a 1,000 kw. *Kerr* Curtis-Rateau type turbine—two Curtis stages and twelve Rateau stages.

about 1,000 and 35,000-kw. capacity are of the composite-staged type. However, they are also regularly manufactured in smaller capacities. Their usual operating speeds are from about 1,500 to 5,000 r.p.m. They are made for high, low

and mixed pressures, condensing and non-condensing, and (Div. 9) for extraction service. In general, they are used to drive large-capacity generators.

65. In A Reaction Turbine The Steam Expands In Both The Moving And In The Stationary Blades (see Sec. 31 and Fig. 77).

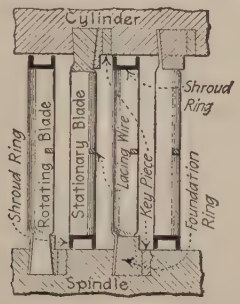


FIG. 77.—Obsolete *Allis-Chalmers* reaction-turbine blading.

The steam is admitted to the first row of nozzle blades (*E*, Fig. 78) at full inlet pressure. The steam, in passing through these blades, undergoes a slight expansion. A further expansion of

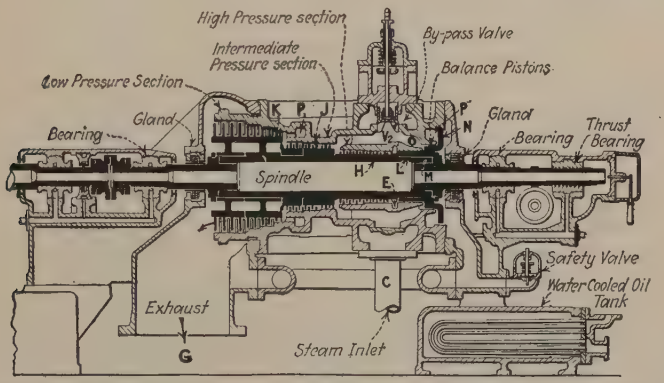


FIG. 78.—Longitudinal section of a single-flow reaction turbine. (*Allis-Chalmers Mfg. Co.*)

the steam occurs in the moving blades; the work of rotation is thus produced by reactive forces (Sec. 7). The action of the steam in each successive reaction stage of a reaction turbine is identical to that in the first reaction stage which is described above. To take care of the increasing volume of the steam

as it expands throughout the successive reaction stages of a reaction turbine, the blade lengths and the rotor diameter are increased by successive steps (barrels, Sec. 25) or sections (*H*, *J*, and *K*, Fig. 78). A reaction turbine is sometimes called the *Parsons* type of turbine.

66. Reaction Turbines Are Manufactured For Nearly All Steam Conditions. However, they are usually designed for operation on high- or low-pressure steam and to exhaust into a high-vacuum condenser. The reason for the general use of a high-vacuum condenser with turbines of this type is that the intermediate and low-pressure sections (Fig. 78) are more efficient than is the high-pressure section. The most efficient speeds of reaction turbines are usually lower than are those of impulse turbines because reaction turbines are generally built with very many more stages. Consequently, they are generally used to drive large alternators through a direct connection, at some speed between about 750 and 3,600 r.p.m. Reaction turbines are made in capacities of from about 300 to 30,000 kw. For capacities above about 30,000 kw., *compound turbines* (Sec. 68) are generally used. Various types of reaction turbines are described in the following sections.

67. The Three Principle Types Of Reaction Turbines (Table 47) are: (1) *Single-flow*, Sec. 44 and Fig. 78. (2) *Double-flow*, Sec. 45 and Fig. 79. (3) *Single-and-double-flow*, Sec. 46 and Fig. 80. Reaction turbines of each of these types are described in the notes below.

Note.—A Single-flow Reaction Turbine is shown in Fig. 78. The live steam is admitted through the inlet, *C*, to the high-pressure section, *H*, of the cylinder at *E*. After passing through the turbine, the steam is exhausted at *G*. The difference in the steam pressure—which is caused by the expansion of the steam in the moving blades, Sec. 65—on the two sides of each row of moving blades produces an end thrust in the direction (to the left in Fig. 78) of the steam flow. To equalize this end thrust, balance pistons, *L*, *M* and *N*, are provided, respectively, for each of the three sections *H*, *J* and *K*. These pistons connect with the high-pressure ends of their respective sections by the passageways *E*, *O* and *P*. The area of the balance pistons, *L*, *M* and *N*, is just sufficient so that the steam pressure on them exactly balances the end thrust to the left. To operate at overload, the governor-controlled bypass valve, V_2 (Sec. 154), admits steam directly to the intermediate-pressure section *J*.

Note.—In A Double-flow Reaction Turbine (Fig. 79), the steam is admitted at the center of the blading at *A*. There the steam divides into two equal parts. One-half of it flows to the left and the other half flows to the right. Consequently the end thrust (see preceding note) in

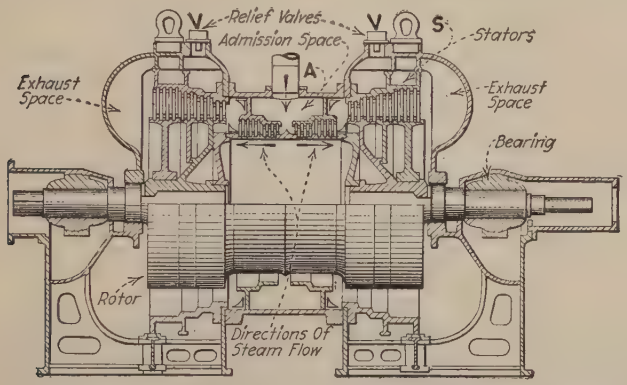


Fig. 79.—Low-pressure double-flow reaction turbine. (*Westinghouse Electric & Mfg. Co.*)

one-half of the turbine is counter-balanced by that in the other half, thus obviating the necessity of balance pistons. Also, since the steam

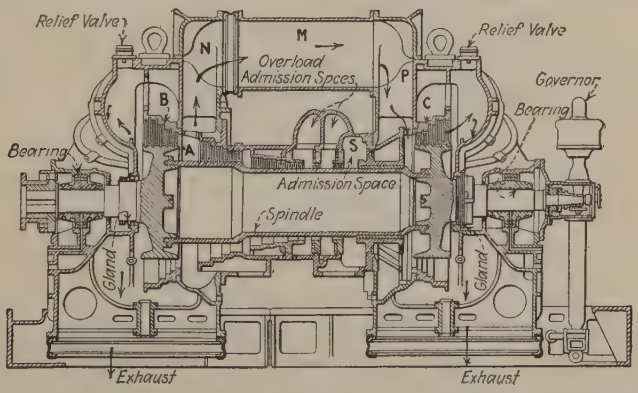


Fig. 80.—Section of a 30,000-kw. single-double-flow steam turbine having reaction blading and complete expansion within a single cylinder. (*Westinghouse Electric & Mfg. Co.*)

is divided into halves, the diameter of the rotor can, in a double-flow turbine, be made smaller than in a single-flow turbine of equal rating.

Note.—A Single-and-double-flow Reaction Turbine (Fig. 80) provides a means of utilizing the energy in the large volume low-pressure

steam without unduly increasing the blade lengths in the low-pressure sections of the cylinder. In Fig. 80, the steam enters the turbine at the admission space, *S*, and flows through the turbine, in a left-hand direction, to *A*. At *A*, the steam divides, one-half flowing through the low-pressure section *B* and the other half, via *NMP*, through the low-pressure section *C*, to the exhaust.

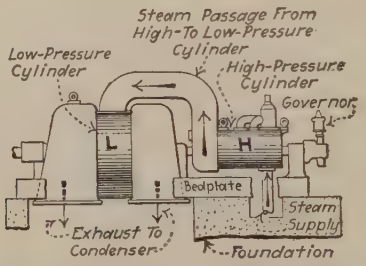


Fig. 81.—Exterior view of a tandem-compound reaction-turbine; the high-pressure cylinder, *H*, is of the single-flow type, the low-pressure cylinder, *L*, of the single-and-double-flow type. See Fig. 245A for a sectional elevation of a tandem-compound reaction turbine.

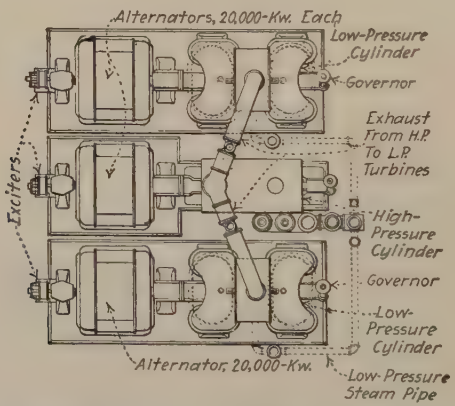


Fig. 82.—Three cylinder cross-compound, 50,000-kw. reaction turbine unit consisting of one high-pressure and two low-pressure elements. (*Westinghouse Electric & Mfg. Co.*)

68. A Compound Steam Turbine is one wherein the total steam expansion from boiler pressure to condenser pressure occurs in two or more separate cylinders. Compound steam turbines are (Table 47) made: (1) *Tandem-compound*, Fig. 81, wherein the axes of both cylinders lie along the same straight line. A tandem-compound turbine unit is usually direct-con-

nected to a single generator. (2) *Cross-compound*, Fig. 82, wherein the axes of all cylinders are not in the same line, but usually in parallel lines. Each element, or cylinder, of a cross-compound turbine unit is usually direct connected to a separate generator. The tandem-compound reaction turbine which is shown in Fig. 81 has a high-pressure cylinder of the single-flow type and a low-pressure cylinder of the single-and-double-flow type.

69. An Impulse-and-reaction Turbine (Fig. 83) is, in addition to the reaction blading, *R*, generally provided with two

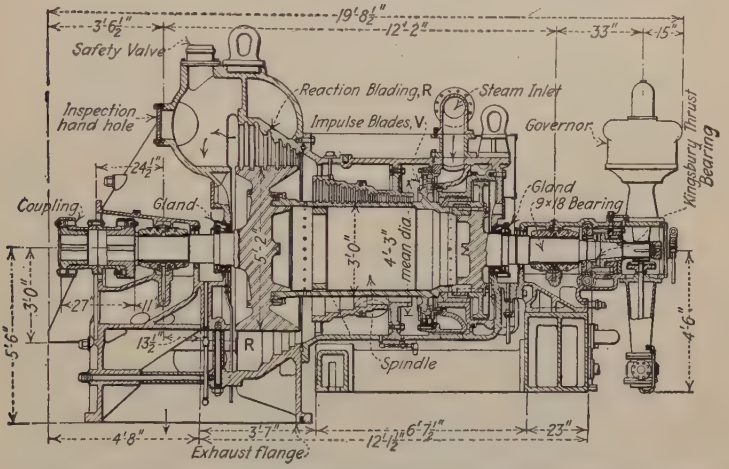


Fig. 83.—Single-flow impulse-and-reaction turbine of 10,000-kw. capacity. (*Westinghouse Electric & Mfg. Co.*)

rows of moving blades, *V*, of the velocity-staged impulse type (Sec. 57). The steam flows through this impulse blading before it reaches the reaction blading. Thus both the temperature and pressure of the steam is decreased before it enters the first reaction stage. Since the steam pressure on the first reaction stages is thereby decreased, the leakage of steam over the ends of the short reaction blades will not be as great as if the high-pressure steam were admitted directly to the first reaction stage as is done in turbines (Sec. 65) of the purely reaction type. Also, since the temperature of the steam is, in the impulse-and-reaction turbine, lowered before it reaches the

reaction blading, the high-pressure section of an impulse-and-reaction turbine is not subjected to such high temperatures as

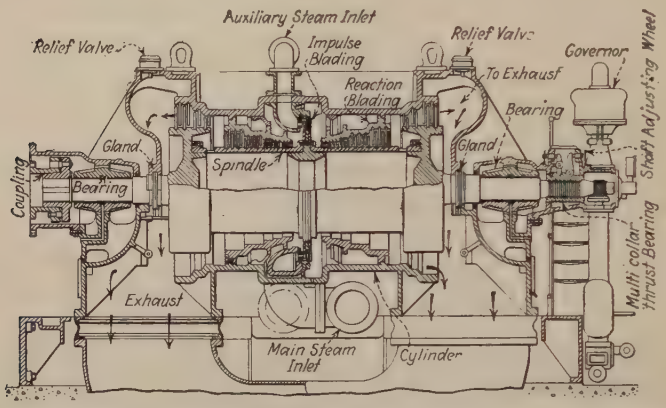


FIG. 84.—Section of a 20,000-kw., 1,500-r.p.m., double-flow *Westinghouse* impulse-and-reaction turbine.

is the high-pressure section of a purely reaction turbine. For the same reasons as were given for reaction turbines (Sec.

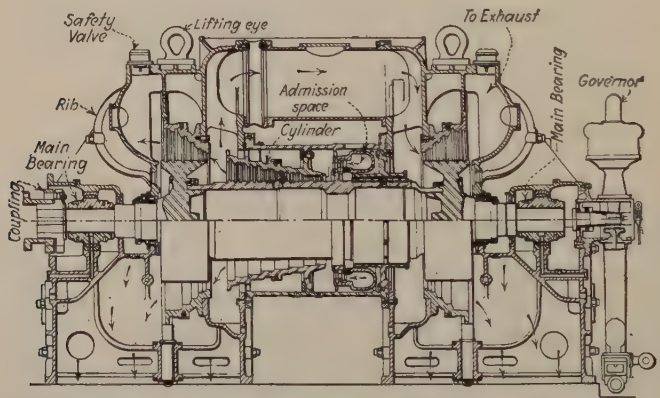


FIG. 85.—*Westinghouse* single-and-double flow, impulse-and-reaction 25,000-kw. turbine.

67), *impulse-and-reaction turbines are made single-flow* (Fig. 83), *double-flow* (Fig. 84), and *single-and-double-flow* (Fig. 85).

QUESTIONS ON DIVISION 3

1. Name the three fundamental types of steam turbines.

2. What is a horizontal turbine? A vertical turbine? What tends to cause vertical turbines to be unreliable?

3. What are the four principal types of impulse steam turbines? What two of these types are frequently combined into one turbine?

4. Name the classification of impulse turbines according to the name of the man obtaining the original patent. Make a sketch to illustrate each.

5. For what purposes are single-stage impulse steam turbines generally used? What is about their usual maximum horsepower rating? Range of pressure ratings? Range of speed ratings?

6. Name two principal types of single-stage impulse steam turbines and explain with a sketch the action of the steam in each type.

7. Name the two principal types of single-stage re-entry turbines and make a sketch to show the path of the steam through each type.

8. For what classes of service are single-stage impulse re-entry turbines especially adapted?

9. Explain with a sketch the action of the steam in a turbine of the velocity-staged type. Does the velocity of the steam with respect to the vanes or blades change in passing through them and if so how?

10. What is the maximum number of rows of moving blades which is generally used in velocity-staged turbines?

11. Why are velocity-staged turbines sometimes used in preference to single-stage turbines?

12. Make a sketch to show the usual arrangement of the nozzles, moving and stationary blades in a velocity-staged turbine.

13. What are the principal applications of velocity-staged turbines? For what speeds, horsepowers and steam conditions are they usually designed?

14. Make a sketch of and explain the action of the steam in a pressure-staged impulse turbine.

15. What is the purpose of pressure staging? Explain how pressure staging accomplishes this purpose. Has pressure staging any advantage over velocity staging and if so what is it?

16. What is a diaphragm? What means are generally employed to minimize steam leakage through the clearance between the diaphragm and the shaft?

17. What two methods are used on pressure-staged turbines to provide for overload?

18. Give the horsepower range, the usual steam conditions and the speed range for which pressure-staged turbines are ordinarily designed.

19. Make a sketch to illustrate the action of the steam in a velocity-and-pressure-staged turbine. What is the purpose of velocity-and-pressure staging?

20. What are the horsepower range, the usual steam conditions and the speed range for which velocity-and-pressure-staged turbines are usually designed?

21. Make a sketch to explain the action of the steam in a composite-staged turbine. What is the reason for using composite staging? Within what horsepower and speed ranges are composite-staged turbines usually designed to operate?

22. Explain the action of the steam in a reaction turbine.

23. Give the range of speed and horsepower ratings for which reaction turbines are ordinarily designed. For what steam conditions are they especially suitable?

24. What are three principal types of reaction turbines?

25. Why are balance pistons generally used in single-flow reaction turbines? Why are they not required in double-flow reaction turbines?

26. Why is the single-and-double-flow construction used in large reaction turbines?

27. What is a compound turbine? What is a tandem-compound turbine? What is a cross-compound turbine?

28. What are the advantages of an impulse-and-reaction turbine over a reaction turbine?

29. Make a complete table showing the classifications of all steam turbines according to general construction.

70. Table by Manufacturers Showing Steam Conditions,

Manufacturer	Fig. No.	Steam conditions	Class or description
Alberger Pump & Condenser Co.	..	High-pressure, non-condensing	Curtis, impulse, single flow
Allis-Chalmers Mfg. Co.....	78	All	Parsons, reaction, single-flow
Carling Turbo-Blower Co.....	64	High-pressure, non-condensing	Impulse, single-flow
Coppus Engineering & Equipment Co.	50	High-pressure, non-condensing	Impulse, single-flow
De Laval Steam Turbine Co....	51	High-pressure, condensing and non-condensing	Impulse, single-flow
	. ..	High-pressure, condensing and non-condensing	Impulse, single-flow
	67	All	Impulse, single-flow
General Electric Co.....	70	High-pressure, non-condensing	Curtis, impulse, single-flow
	72	All	Curtis, impulse, single-flow
Ingersoll-Rand Co.....	73	High- and mixed-pressure, condensing	Impulse, single-flow, and double-flow
Kerr Turbine Co.....	 76	All	Impulse, single-flow
Midwest Engine Co.....	52	High-pressure, non-condensing.	Impulse, re-entry
Moore Steam Turbine Corp.....	 75	High-pressure, condensing and non-condensing	Impulse, single-flow
Parsons Marine Steam Turbine Co.	.....	High- and low-pressure	Parsons, reversing

* Oil relay governors used on some large machines.

† Steam relay governors made at Lynn works; oil at Schenectady.

Classes and Approximate Ratings of Steam Turbines

Approximate ratings	Type of staging	Governor	Glands	Notes
10–50 hp.	Velocity	Pickering	Lantern	Centrifugal-pump drive
300 kw. and up	Reaction	Throttling and by-pass	Water packed	Turbo-generator, direct drive
1–25 hp.	Velocity	Direct throttling	Metal packing	Blower drive
2–50 hp.	Single	Pickering	Stuffing box	Blower and pump drive
1–600 hp.	Single	Direct throttling	Metal packing	
1–600 hp.	Velocity	Direct throttling	Metal packing	Direct or gear-connected for pump or generator drive
50–15,000 hp.	Pressure	Direct throttling* or oil relay	Carbon packed	
10–400 hp.	Pressure-and velocity	Direct throttling	Steam packed	Mechanical drive
100 kw. and up	Composite	Direct,† steam, oil relay	Steam labyrinth	Direct or gear-connected for generator drive
300–4,000 hp.	Composite	Oil relay	Steam labyrinth	Turbo-compressors
Up to 4,500 hp.	Velocity / Composite	Direct throttling* and oil relay	Carbon packed	Direct and gear-connected for pump and generator drive
1–800 hp.	Single	Direct throttling	Carbon packed	Direct connected for pump, generator, and blower drive
Up to 4,500 hp.	Velocity / Composite	Direct throttling and oil* relay	Carbon packed	Direct and gear-connected for pump, generator and blower drive
.....	Reaction	.....	.....	Marine service

70. Table by Manufacturers Showing Steam Conditions, Clas-

Manufacture	Fig. No.	Steam conditions	Class or description
Power Turbo Blower Co.....	48	High-pressure, non-condensing	Impulse, single-flow
Ridgeway Dynamo & Engine Co.	69	High- and mixed-pressure	Impulse, single-flow
Steam Motors Co.....	65	High-pressure, non-condensing	Impulse, single-flow
B. F. Sturtevant Co.....	31	High-pressure, non-condensing	Impulse, tangential-flow
Terry Steam Turbine Co.....	57	High-pressure, non-condensing	Impulse, tangential-flow
	74	High-pressure, condensing	Impulse, single-flow
Westinghouse Electric & Mfg. Co.	53	High-pressure, non-condensing	Impulse, re-entry
	80, 82, 83, 84, 85	All	Single-flow, double-flow, single-and-double flow, single-cylinder and compound
L. J. Wing Mfg. Co.....	..	High-pressure, non-condensing	Impulse

• Oil relay governors used on some large machines.

ses, and Approximate Ratings of Steam Turbines—*Continued*

Approximate ratings	Type of staging	Governor	Glands	Notes
1–20 hp.	Single	None	Metal packing	Blower drive
500–5,000 kw.	Composite	Pneumatic oil relay	Carbon packing	Generator drive
Up to 300 hp.	Velocity	Direct throttling	Steam labyrinth	Direct-connected pump drive
5–350 hp.	Single	Direct throttling	Stuffing box	Direct pump and generator drive
15–500 kw.	Single	Direct throttling	Carbon packed	Direct and gear-connected for pump, blower and generator drive
1–1,500 hp.	Composite	Direct throttling or oil relay*	Carbon packed	
2–640 hp.	Single	Direct throttling	Water seal	Mechanical drive
500 kw. and up	Reaction, impulse-and-reaction	Steam or oil relay, and intermittent	Centrifugal water-packed	Turbo-generators
1–20 hp.	Single	None	Stuffing box	Blower drive

DIVISION 4

STEAM-TURBINE INSTALLATION

71. The Various Steps In Installing A Steam Turbine may be given in the order in which they should usually be performed as follows: (1) *Plan* if necessary. The principal parts of the installation which must be planned are the foundation and piping. (2) *Build the foundation.* (3) *Receive and unpack* the turbine. (4) *Place the turbine on the foundation, level and grout.* (5) *Make initial adjustments.* The bearings, coupling, governor and thrust bearings should be adjusted sufficiently so that the turbine may be turned over slowly without damage. (6) *Connect up* the condenser, oil system, piping, drains, and other accessories. (7) *Make final adjustment* under operating conditions. (8) *Start up* the first time. The governor must be adjusted by running the turbine at its rated speed.

72. In Planning The Installation Of Large Turbines, (Fig. 86) provision should be made for the space and support required by all principal piping, bearing in mind that the turbine casing must not be subjected to piping strains. The location of all auxiliaries and accessories should be carefully planned so that they can be readily handled by the crane and so that they are all as nearly alike as possible thus facilitating the stocking of spare parts. The method of cooling the generator, of supporting the condenser and of connecting the turbine exhaust passage should be completely planned. The planning of large turbine foundations and supports involves provision for the extra stresses occasioned by the the vacuum in turbine casing. (See the author's MACHINERY FOUNDATIONS AND ERECTION.)

NOTE.—THE PIPING FOR A SMALL TURBINE (Fig. 87) need not ordinarily be accurately planned. The turbine may be located where desired and a pipe line run to it and exhaust line run from it by an experienced steamfitter but it must then be properly supported to relieve the turbine

casing of all stresses. Provision is sometimes made in small turbine piping for special governing. Pressure-controlled diaphragm valves are sometimes used on turbo-blowers for boiler furnaces (Fig. 87) so that the speed of the turbine will be proportional to the steam requirements of

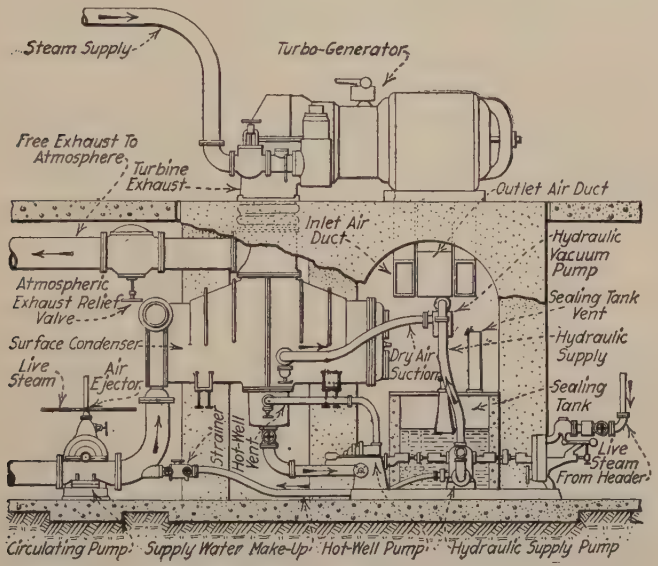


FIG. 86.—Turbo-generator installation showing principal auxiliaries and piping.

the plant—if the boiler pressure falls, the turbine will furnish more air and vice versa. Similar valves or pump governors are sometimes used on turbine-driven boiler-feed pumps to keep a desired water pressure in the feed line.

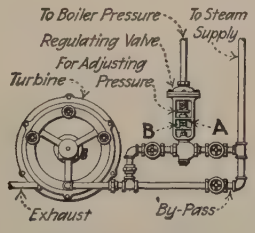


FIG. 87.—Piping connections, for turbine-driven blower, which enable blower to maintain a constant steam pressure in the boiler which it serves.

73. Foundations For Large Or Medium-sized Steam Turbines are ordinarily built in hollow form so that the condenser and other auxiliaries may be placed directly beneath

the turbines. The foundation is frequently in the form of a raised platform or capslab supported from a subbase or footing by a superstructure consisting mainly of vertical columns. Foundations for small non-condensing turbines are simple blocks of concrete which differ in no essential respect from foundations for small motors or other small machinery. In no event, however, should a turbine bedplate be bolted to a wooden floor without building for it a suitable rigid concrete slab (or structural steel frame for small turbines) which will protect the bedplate from possible bending. It should be remembered that the function of the foundation is to maintain the unit in alignment. Permanence of alignment is largely dependent upon the rigidity of the foundation.

74. In Receiving And Unpacking A Turbine Which Is Shipped Assembled (from *General Electric Co.* "Instruction Book 82,200") see that the blanks over the piping outlets and inlets are intact and that no foreign material has lodged in the steam passages. Look over the gages and piping and see that all the fittings are in place. Report any shortages as soon as possible. When ready to install the machine, wipe off all slush with clean waste and, if carbon packing boxes or other machined surfaces coming in contact with the steam or exposed to view or touch are slushed, they should be cleaned thoroughly with gasoline. No bearings, linings, journals, or roller or ball thrusts should be cleaned with gasoline but merely wiped clean with waste.

75. In Receiving And Unpacking A Turbine Which Is Disassembled, locate all parts called for on the shipping memorandum. If the parts are to be assembled immediately, wipe off all slush or clean with gasoline as previously noted. The wheels and shaft will, in most cases, be shipped resting in blocks fitted to the recesses in the heads where the carbon boxes belong and this blocking should not be removed until the wheel casing is resting in the base and the shaft in the linings. See that all the blanks over the openings are intact and that no foreign material has found its way into any part of the machine.

76. Turbines Are Placed On Their Foundations And Aligned On Wedges (Fig. 88). The wedges are of steel about

1 in. thick. The primary alignment and leveling is usually made with all principal parts of a turbine in place but before the piping and auxiliaries are connected. The machine is slid slightly or the wedges driven in or out until the desired level of the bedplate is obtained. The level is indicated by placing an accurate spirit level across the finish bosses of the bedplate. These bosses are usually provided at convenient points on the bedplate and are scraped to an accurate level at the factory. It is not sufficient to try the level at one or two

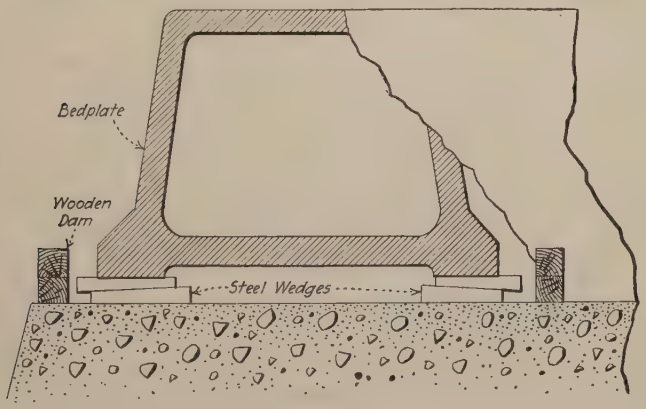


Fig. 88.—Bedplate of turbine supported on wedges and surrounded by wooden dam for grouting.

points. It should be tried all the way around since there is often some warping of the bedplate in shipment. The bedplate is then grouted to the foundation by pouring thin grout or cement mortar under the plate. A dam is built (usually of strips of wood) to confine the grout and force it to flow under the plate and up inside for 2 or 3 in. After about 2 or 3 hr. the dam is removed and the excess grout trimmed off. About two days later, the wedges may be removed, if desired, and the anchor bolts tightened. See the author's Machinery Foundations And Erection for further information.

77. In Handling Small Turbine-driven Sets (sizes up to about 100 kw.) which are usually shipped completely assembled, no unusual amount of care is necessary. In general they can be rolled on skids without special regard to deflecting

the bedplate; or, they can be picked up by a crane with almost any convenient hitch without fear of undue buckling. They are thus readily set on the foundation where the bedplate may be leveled by means of supporting wedges (Sec. 76) although very careful leveling is not usually necessary.

78. In Handling Medium-sized Turbine-driven Sets (150 to 2,000 kw.) the bedplates should be given uniform support to insure against deflection (bending) by the heavy supported parts. These machines are usually shipped assembled except for small parts. If rolled on the skids, great care should be exercised to see that the skids are supported at a number of points. Likewise, when hitched to a crane hook, the heavy parts should be supported individually instead of being carried on an unsupported part of the bedplate. When set on the wedges on the foundation top, the bedplates should be very carefully leveled by means of the spots or surfaces provided therefor; see the author's MACHINERY FOUNDATIONS AND ERECTION.

NOTE.—A MEDIUM-SIZED TURBINE-DRIVEN UNIT MAY BE ALIGNED AT ITS COUPLING; see Sec. 167 for the method. When so aligned, however, account must be taken of the fact that, after the turbine-end of the unit is heated by the steam which it contains when operating, it will have expanded and will stand at a higher elevation than when cold. Allowance must be made for the amount that the turbine end will rise; see Sec. 85.

79. In Erecting Large Turbo-generator Sets it is important to plan the work as completely as possible so that the erection will progress smoothly and that the man in charge can give his entire attention to the work at hand without fear of getting "hung up" or wasting time and labor. The following procedure (based on an article by E. H. Thompson in *Power*, July 6, 1920) will be of value in such work:

1. WHEN THE MACHINE IS RECEIVED on freight cars, the various parts must be identified and arrangements made for unloading. It is necessary to consider which parts are first needed, which are to be stored until later, where and how these are to be stored, and how transported to the foundation when needed. In most plants the cars are brought within reach of the power-house crane; often it is necessary to roll or

drag them to the crane. Sometimes the plant may be in the process of building with no crane in operation, then rigging work is the largest problem.

2. The Shipping Lists Can Be Checked as the unloading proceeds. Meanwhile a shack can, if necessary, be erected near the installation for tools, storing delicate parts, blueprints, and for the convenience of the men. Wedges and blocks for the grouting and special tools can be ordered for the work, to be ready when needed.

3. The Bedplate Is First Placed On The Foundation.—A sectional baseplate should be assembled by either heating the bolts or driving the wrench with a sledge. The bedplate should then be checked for accuracy. The bedplate may then be located on the foundation according to the center lines shown on the plant-design drawing. All openings in the foundation—for pipe connections, generator air ducts, drains and the like—should be checked for accuracy. Sometimes it is well to check openings and connections by temporarily assembling parts of the turbine casing or generator. A little such forethought may obviate the necessity of moving a 100-ton condenser or of chipping a concrete opening at the last minute, or of straining pipe flanges to make connections and causing a bad joint, or other trouble. The bedplate can then be carefully leveled.

4. The Bearing Pedestals, Turbine Casings, Generator And Other Parts Which Must Be Aligned may now be placed on the bedplate. A steel wire is generally used for aligning. The end bearings are first carefully doweled and bolted into their permanent position. A *new* steel wire 0.008 to 0.010 in. in diameter, such as piano wire, is tested to breaking strain by lifting various weights with it. The line is then stretched between two rigid supports, such as heavy timbers or convenient columns or pieces of machinery and a tension is produced in it by suspending from it a weight of about ¾ of its breaking load.

The Line Is Now Moved Up Or Down Or Crosswise at each end until it is exactly central with the bored surfaces of the end-bearing pedestals or other parts used as a permanent guide. Wedges to suit the rigging are convenient in making small changes in position of the wire. The distance from the wire to the bored surface can be roughly measured with an inside caliper and with final accuracy by an inside micrometer or pin gage. The pin gage is generally the machinist's choice and is made by selecting a piece of wood ¼ to ¾ in. in diameter, and ¾ in. shorter than the average measurement. A pin or needle is driven in at each end so that measuring is done between the two pinheads or needles. The distance is changed by driving the pins in or pulling them out. The position of the wire must be adjusted so that the radial distance to the bored surface is the same at each side as well as above and below. It is not difficult to obtain an accuracy of 0.000,5 in.

After The Tight Line Has Been Set, the other bearing pedestals, turbine casings, generator, gear housing, etc., can be adjusted so as to be central with the line. They should then be doweled, by drilling and reaming and accurately fitting dowels.

The Middle Bearing Will Be Lower Than The Outside Bearings by an amount equal to the sag in the wire. This would ordinarily be negligible up to 15-ft. span of wire. It can be checked by assembling the rotary element and opening up the coupling; see Sec. 167. The middle bearing should be shimmed until the distance between the flanges of the coupling is the same above as it is below, or perhaps 0.002 to 0.006 in. more on top.

The Turbine Casing Is Usually Set Low to allow for expansion. It is best, if any question arises in this connection, to get the manufacturer's information on this point. It can readily be settled, however, by making the adjustment that seems best, and then checking the alignment with shaft again, when the machine has been put in service, and correcting to be central under working conditions of temperature. If the shaft is sprung, the amount of the "spring" must be measured, and this should be taken into consideration when centralizing with the shaft.

5. The Grout Should Be Poured under the bedplate after the foregoing questions of checking location, alignment, pipe connections, etc., have been settled. While this is hardening, the parts next to be assembled should be cleaned and made ready.

The Importance Of Cleaning is seldom realized by the inexperienced. The practical man knows that grit or sand in a bearing running at high speed can cause considerable damage in a few minutes. A scraped joint, where no gasket is used, is difficult or impossible to make steam-tight when dirt is present. Dirt causes alignment troubles when shims and contact surfaces are not properly cleaned. Cleaning is something of an art. An appreciable skill is required in preparing, with a sharp machinist's scraper, a scraped surface which has been warped by bolt pressure and expansion, and which is covered with sticky dope. Some judgment is required in getting this work done by unskilled and unreliable labor. When large pieces are cleaned, it is a good plan to provide putty knives or old files ground to a dull edge. The work can then be inspected by an experienced man, and filing or scraping done as required.

Delicate Parts Which Are Being Assembled Should Be Covered With A Tarpaulin Or Other Shield each night to prevent dust settling from the air, and solid particles, such as bolts or nuts, finding their way in. The steam passages must be continually guarded and inspected.

The Most Difficult Part of The Assembling Is Usually The Turbine Rotor And Casing. It is most important to have reliable men to watch different parts as the lowering is done.

The Remaining Parts To Be Assembled, which include generator, packing casings, steam chest, valve gear piping, etc., often require much painstaking work and represent a large part of the job.

80. Casings Of Long Horizontal Turbines Are Usually Bolted Down At One End Only. Due to the difference in length of a long casing when hot and when cold, it is necessary that one

end be allowed to slide freely. The *General Electric Co.* gives the following directions in connection with the installation of their 12-stage Curtis turbines: The bolts holding down the standard at the high-pressure end of the machine should not be drawn up so tight as to prevent relative movement of standard and base at this point. The turbine casing is doweled to the base at a point approximately near the center of the exhaust passage, and expansion due to temperature changes will cause a movement of the standard relative to the base. Alignment is preserved by keys. Marks should be placed on both standard and base to see that this movement actually takes place.

81. To Compensate For Expansion And Wear Of Bearings, shims which are provided for the purpose by the manufacturers should be placed under the bearing pedestals. No shims should be used between the turbine casing feet and the supporting pads of the bedplate. *Insulating shims* are sometimes necessary under the generator end bearing; see Sec. 200. Tests are made at the factory to determine if these shims are necessary. If so, they are always furnished with the machine. The bearings are aligned by means of a tight line stretched through the assembled shells of the bearings. The turbine end bearing must be aligned with special accuracy because the worm gear drive for the governor will not operate satisfactorily if there is any misalignment at this point.

82. When A Turbine Is Shipped Entirely Disassembled, the bearings may be aligned by means of a fine steel wire tightly stretched through the bearing center line as explained in Sec. 79. When the bearings themselves are received disassembled, they should be examined and flushed out with kerosene before assembling. They should, after the primary assembly and alignment have been made, be filled with the proper grade of oil. The *cooling coils* of the oil systems should be inspected for leaks by applying the full water pressure before the oil system is filled. Leakage of water into the oil causes much trouble. The oil system should then be cleaned if necessary, filled and examined for leaks; see also Sec. 204.

83. The Axial Blade Clearance Of Turbines May Sometimes Be Tested By Means Of A Taper Gage. A plug hole, *H*, Fig.

89 is usually provided for such testing in each pressure stage of most impulse turbines. To make the test, remove the plug and insert the taper gage on each side of the fixed blades as shown in Fig. 89. The clearance should be the same on each side of the fixed blades. If it is not, the difference should be corrected by adjusting the thrust bearing as explained in Div. 5. If the condition is different in two or more stages, the adjustment should be made so that the least clearance in any stage will be as large as possible. *If there is no plug hole for measuring the axial clearance,* the adjustment may be made by adjusting the rotor first to one extreme position and

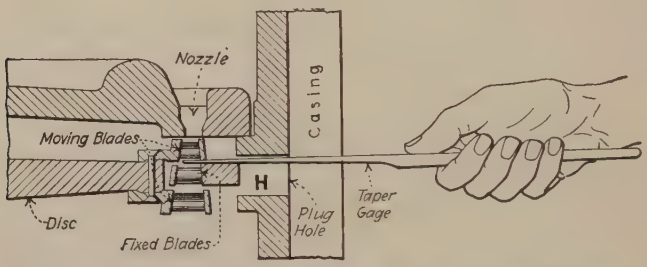


Fig. 89.—Method of checking the axial clearances of a Curtis turbine.

then the other. By carefully turning the rotor and listening for sounds of interference the extreme positions may be ascertained but this must be done very cautiously to avoid damage. The extreme positions may be marked on the shaft. See Div. 5 for further instruction. The axial alignment of some bucket-wheel turbines may be tested as shown in Fig. 60 by removing the plug provided for the purpose and observing the alignment.

84. Some Miscellaneous Precautions Which Should Be Taken In The Piping Of Large- Or Medium-sized Turbines are as follows:

Piping to the turbine should be as short as possible, should be of ample size to prevent excessive pressure drop, should be formed in smooth bends whenever possible, should be so shaped that expansion will not strain it, and so supported that it will not bear heavily on the turbine casing. Cut-outs or stop valves should be provided in the branch leading from the main header to the turbine so that the whole pipe will not fill with water by condensation when the turbine unit is idle. Separators

should always be provided, where saturated steam is to be used, in the piping just before the steam is admitted to the turbine. Where superheated steam is to be used, the use of a separator is unnecessary—provided that the superheat is not lost by radiation in the piping and provided also that precautions are taken to prevent the flow of condensed steam into the turbine when starting. Before the piping is connected to a turbine, the live steam should be blown through it to remove dirt and scale.

Strainers must always be used on high-pressure turbines and should be removable for cleaning. Strainers are usually provided by the turbine manufacturer just ahead of the governor valve. If none is provided, one should be procured and inserted. For low-pressure turbines using steam through a separator from a reciprocating engine, the strainer is sometimes omitted.

Drains should be provided to take the drips from the throttle valve, separator, and exhaust end of the turbine casing, and low points in the piping where water is likely to collect. These drains may usually be combined and run to the condenser. A valve must be provided at the head of each drain to close it off as soon as all the water is removed. Where the condenser is located too high to take the drains, a trap should be provided which will deliver the drips to the hotwell.

Casings Should Be Protected From Piping Strains and all other kinds of strains. The capslab (supporting slab) of the foundation should be so rigid that no deformation is possible. The grouting of the bedplate to the slab should be so thorough that no uneven support is formed which will cause warping. The condenser connections and other low-pressure steam connections should (unless a spring-supported condenser is provided) be made with expansion joints so that no strain will be transmitted from the condenser or other structure to the casing; see the author's Machinery Foundations And Erection. The relief valves on the casing and the atmospheric relief valve on the condenser connection should be in good condition to avoid straining the casing or shutting down of the unit in case of a condenser failure. The relief valve should be set for about 2 lb. per sq. in. gage. There should be no hand-operated valve which can prevent the steam escaping through the relief valve.

85. The Final Alignment Of Turbine-driven Units On Their Bedplates Or Soleplates Is Preferably Made When The Unit Is At Operating Temperature.—The steam end of the unit expands when heated and, if aligned while cold, will not run true unless allowance is made for the expansion. The steam end of the unit should, for condensing operation, when cold, be lower than the generator end by about 0.005 in. per ft. of vertical distance from the point where the casing is supported to the shaft center and 0.01 in. lower for non-condensing

operation. As turbine-driven boiler-feed pumps are almost as hot on one end as the other, very little allowance need be made for their expansion.

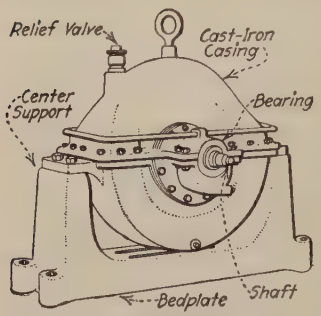


FIG. 90.—Small *Westinghouse* turbine so supported that changes in temperature will have little effect on the shaft alignment.

NOTE.—SOME SMALL-TURBINE CASINGS ARE SUPPORTED AT THE HEIGHT OF THEIR SHAFTS (Fig. 90) so that increased or decreased temperature will have little or no effect on the alignment.

86. The Governor And Its Operating Mechanism Should Be Inspected to make sure that it has not been damaged in shipment. If necessary, the governor should then be adjusted to a preliminary setting (see Div. 6). If a generating unit which is being installed is to operate in parallel with units already installed, the governor should be carefully adjusted for the same speed regulation as the other units (see Div. 6 and the author's STEAM-ENGINE PRINCIPLES AND PRACTICE). The amount of speed variation obtainable with the synchronizing mechanism should be noted and adjusted if necessary to permit synchronizing.

87. Instructions For Checking Alignment, are given by the General Electric Co. as follows for their one- and three-stage turbine alternators: *Bosses for checking the axial alignment in a horizontal plane* will be found on or close to the horizontal center-lines of the unit. One of the bosses is located on the wheel casing, close to the outboard end. A second boss is located on the connection piece adjacent to the stator. A third boss is located on the generator outboard-bearing bracket. The two outer bosses are tapped for studs to carry a tight line. A 0.016,6-in. piano wire weighted by a 30-lb. weight and supported at its weighted end by a small roller carried between collars is stretched along the unit. The horizontal distances from the tight line to the bosses on the connection-piece, wheel case and generator bearing-bracket are stamped on these bosses.

To check this alignment, it is necessary to duplicate the distances to the outer bosses and to compare the observed distance to the middle boss with the figures stamped on it. No correction for sag of wire is necessary, but alignment must be checked when the unit is cold.

NOTE.—THE ALIGNMENT IN A VERTICAL PLANE IS CHECKED WITH THE SAME WIRE stretched in the same way across the proper bosses. One boss for this alignment will be found on the outboard end of the wheel casing, between the two bosses used for leveling; a second boss is located on the generator end bearing bracket. One of the two bosses on the connection piece used for the leveling serves as the third boss.

QUESTIONS ON DIVISION 4

1. Give in order the various steps in installing a turbine.

2. Why must the foundation of a large turbine be completely planned?

3. What is the general form of foundation most frequently used for large condensing turbines? For small non-condensing turbines?

4. Name several things which should be done in unpacking a turbine which is received for immediate installation.

5. How high above a foundation should a turbine bedplate be supported for grouting? How soon after pouring should the excess grout be trimmed off?

6. Why are long horizontal turbine casings securely bolted down at one end only? How is the alignment of the other end preserved?

7. How may the cooling coils of an oil system be inspected for leaks?

8. Explain a method of testing the axial blade clearance of a turbine.

9. When is it unnecessary to provide a strainer for a steam turbine? When is a separator unnecessary?

10. Where should drain pipes be provided in and around a steam turbine? Where should the drains lead to?

11. How does an atmospheric relief valve protect a turbine casing?

12. Why should a turbine preferably be finally aligned at operating temperature?

DIVISION 5

STEAM-TURBINE SHAFTS, BEARINGS, AND PACKING GLANDS

88. The Satisfactory Operation Of A Steam Turbine Depends Largely On The Condition Of The Shaft, Bearings And Packing Glands. The operator is not particularly concerned with the turbine shaft except insofar as correct alignment (Div. 7) is concerned. This is because the shaft is designed and made by the manufacturer. It, if properly designed and made, requires practically no maintenance, except for maintaining proper alignment, and is not subject to operating difficulties. Consequently, only the more usual types of shafts are briefly described in Sec. 89. Bearings and the packing glands however, may require considerable attention and maintenance on the part of the operator if the turbine is to operate satisfactorily, the bearings and the glands must be kept by him in the best possible condition.

89. Turbine Shafts, Which Represent Typical Construction as employed by different manufacturers, are shown in Figs.

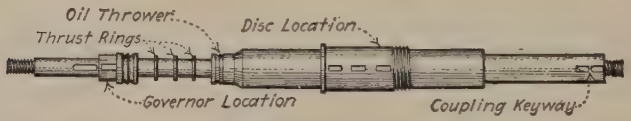


FIG. 91.—Shaft of a *De Laval* turbine showing keyways for fastening the discs and couplings.

91, 92 and 93. The shafts of impulse turbines (Fig. 91) are nearly always made solid, while those of reaction turbines (Figs. 92 and 93) are generally hollow. The shafts of practically all turbines are now made "stiff." See note below.

NOTE.—SOME MANUFACTURERS APPLY THE TERM "SPINDLE" to designate the complete rotating element, as in Figs. 92 and 93. However, the terms "shaft" and "spindle" are generally synonymous.

Note.—The "Critical Speed" Of A Shaft which carries a load, as for instance a turbine rotor, is the specific speed at which the shaft vibrates most violently. If the shaft is permitted to rotate for any length of time at its critical speed, the vibrations may prove disastrous. The explanation for this vibration is too technical to be given here. It

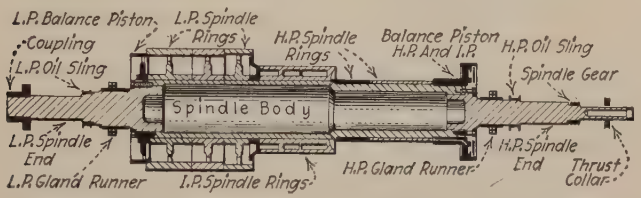


Fig. 92.—Section through the spindle of a *Allis-Chalmers* reaction turbine. (L.P. = low pressure. I.P. = intermediate pressure. H.P. = high pressure.) See also Fig. 93.

is a fact, however, that at speeds well above or well below their critical speeds, all shafts (unless badly unbalanced) will run fairly free from vibration. In the early days of steam-turbine engineering most turbines were operated above their shafts' critical speeds. In starting or stopping such turbines it was essential that the critical speed be passed as quickly as possible. Nowadays, however, nearly all turbine shafts are

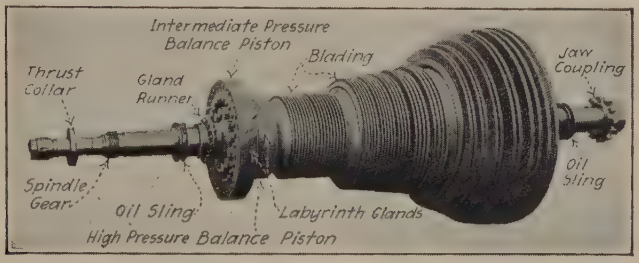


Fig. 93.—Rotor or spindle of a reaction turbine. (*Allis-Chalmers Mfg. Co.*)

designed to be so "stiff" that the turbines operate normally at speeds well below the critical speeds of their shafts. Furthermore, the rotors of modern turbines (the better ones) are carefully balanced in the manufacturers' shops to further lessen the dangers due to vibration.

Note.—A Turbine Shaft Is Said To Be "Stiff" If It Is Designed To Operate At Some Speed Below Its Critical Speed. A turbine shaft is said to be "flexible" if it is designed to operate above its critical speed.

90. The Two Principal Types Of Bearings In Steam Turbines are: (1) *The main bearings*, which carry the weight of

the rotor and which also prevent any excessive movement of the rotor in any direction perpendicular to its axis; the main bearings are sometimes called *radial pressure bearings.* (2) *The thrust bearings* which restrain the rotor from excessive movement in either direction parallel to the axis of the rotor. Bearings of each of these types are discussed in the following sections.

91. Table Showing Classification Of Steam-turbine Bearings. (Only those bearings are included in this table which restrain the movement of the rotor. The bearings which are used in connection with the governor, the oil pump, or other subsidiary apparatus are not included in this table and are not discussed in this division.)

Steam-turbine bearings	Main bearings	Plain bearings	Flexible, Fig. 94
			Rigid, Fig. 100
		Ball bearings	One row, Fig. 50
			Two rows, Fig. 65
	Thrust bearings	Roller, Fig. 103	
		Simple collar, Fig. 104	
		Multi-collar, or marine, Fig. 105	
		Ball, Fig. 106	
		Kingsbury, Fig. 108	

92. Plain Flexible Steam-turbine Bearings (Table 91 and Fig. 94; see also the note below) generally consist of: (1) *The babbitt, B,* which contains the oil grooves, *G,* and upon which the journal bears. (2) *The lining, L,* which is held in place by the spherical seat, *S.* The lining is usually split along the horizontal center line; thus, it is divided into two parts, which are sometimes called the *upper lining* and the *lower lining.* (3) *The pedestal, P,* which supports the lining

through the seat. (4) *The bearing cover, C.* Various manufacturers employ different constructional details in flexible

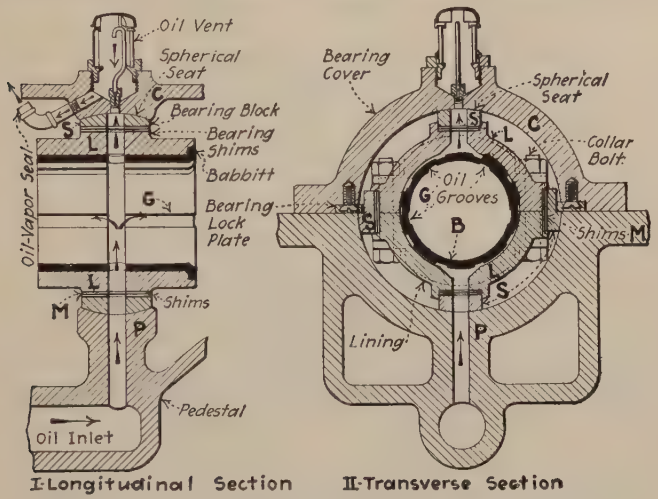


FIG. 94.—Spherical-seated bearing of *Allis-Chalmers* steam turbine.

bearings. This is evident from a comparison of Figs. 94, 95, 96, 97, and 98. Flexible bearings of some kind are used in

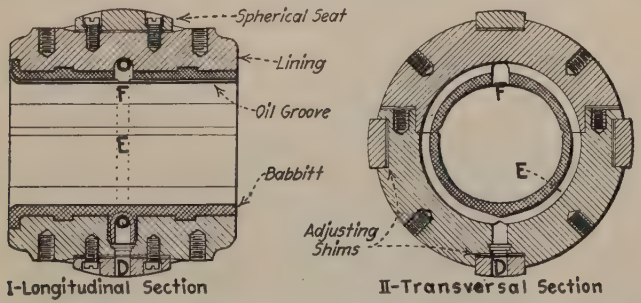


FIG. 95.—Spherical-seated steam-turbine main-bearing. (Oil enters at *D* and passes upward through the spaces *E*, entering the bearing through the groove at *F*. The babbit is so bored that the horizontal "diameter" dimension is slightly greater than the vertical "diameter" dimension. (*Westinghouse Electric & Mfg. Co.*)

nearly all steam turbines. Bearings of this type are also called *spherical-seated bearings*, and *self-adjusting bearings*. The function of a flexible bearing is explained in the following note.

Note.—A Flexible Bearing So Operates that the bearing will, without causing excessive friction, automatically adjust itself so that

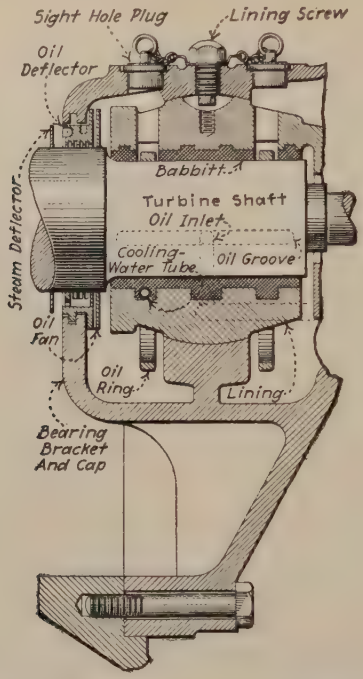


Fig. 96.—Section through outboard bearing of *General Electric Co.* steam turbine.

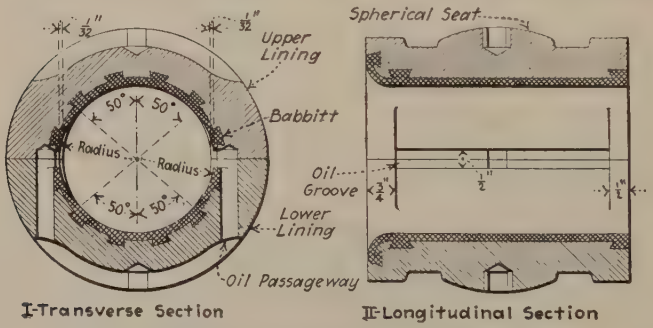


Fig. 97.—Section through bearings of a *Kerr* turbo-generator.

the axis of the bearing coincides with or remains parallel to the axis of the journal or shaft. See Fig. 99. The axis of the shaft when in its normal position is indicated by the center line, *A*. If from any cause,

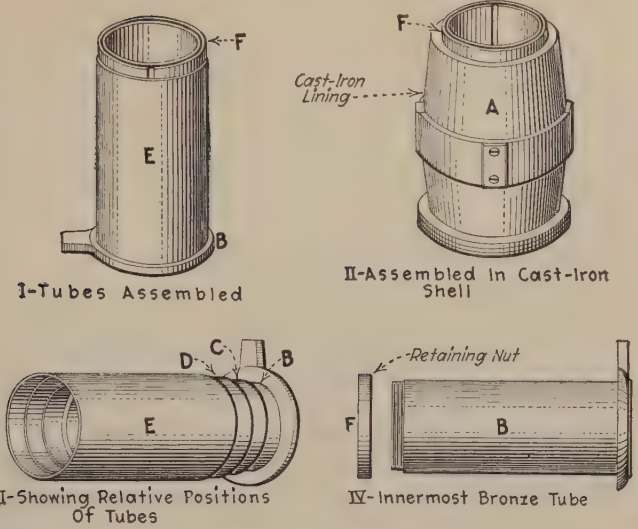


Fig. 98.—Self-adjustable or flexible main bearing consisting of a nest of tubes for high-speed turbines of small capacity. [The bronze tubes, *E*, *D*, and *C* (III) fit over each other (II) with some clearance, so that the innermost is free to move slightly in any direction. Oil fills the clearance between the tubes and forms a cushion which tends to dampen vibration.]

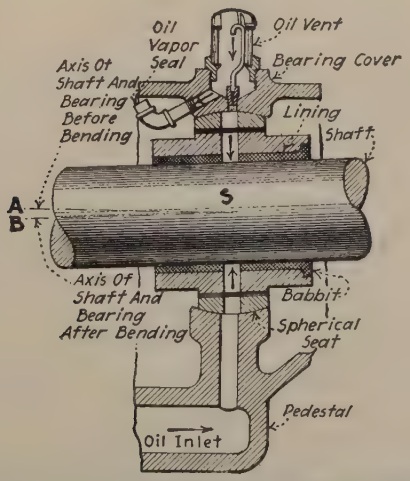


Fig. 99.—Illustrating action of a spherical-seated "flexible" bearing, when the turbine shaft bends. The bending is exaggerated for purpose of illustration.

the shaft should bend, its axis will then be at some other position as indicated by *B*. Now, since the bearing lining is held in a spherical seat (*S*, Fig. 94), the bearing will rotate in this spherical seat and assume the position shown in Fig. 99; thus, coinciding with the axis of the shaft. If the bearing were a *rigid bearing* (Fig. 100) the bearing could not readily adjust itself to any bending of the turbine shaft. Consequently, if the turbine shaft should bend, excessive friction would result and the bearing would be subjected to excessive wear, and probably to heating.

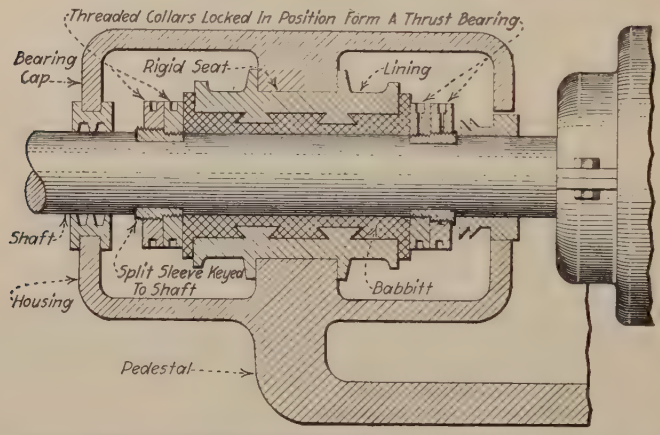


Fig. 100.—Rigidly seated steam-turbine bearing. (Any bending or deflection of the shaft will tend to distort the housing and pedestal, thus causing excessive bearing friction.)

93. Ball Bearings Are Used As The Main Bearings Of Small-capacity Turbines by some manufacturers. A double-race, self-adjusting ball bearing is shown in Fig. 65. The advantages claimed for main bearings of this type are that they minimize friction and are readily accessible for renewal. Ball bearings are seldom used for turbines of capacities greater than about 200 hp. A ball bearing should be flushed out occasionally with kerosene. A ball bearing cannot be repaired; if it becomes badly worn, it must be renewed.

Note.—The Relative Location And The Constructional Arrangement Of Steam-turbine Main Bearings are shown in various illustrations in Div. 3.

94. In General, The Temperature Of The Oil Leaving A Turbine Main Bearing should not exceed about 150° F.

See also Div. 10. However, there are some turbines the bearings of which are designed to operate at a temperature of from 195° to 212° F. To prevent excessively-high bearing temperatures, the main bearings of some medium- and large-capacity turbines are (Fig. 101) water cooled. Cold water is forced through the coils which are imbedded in the bearing lining. Those turbines which have circulation lubricating systems (Div. 10) are generally equipped with separate

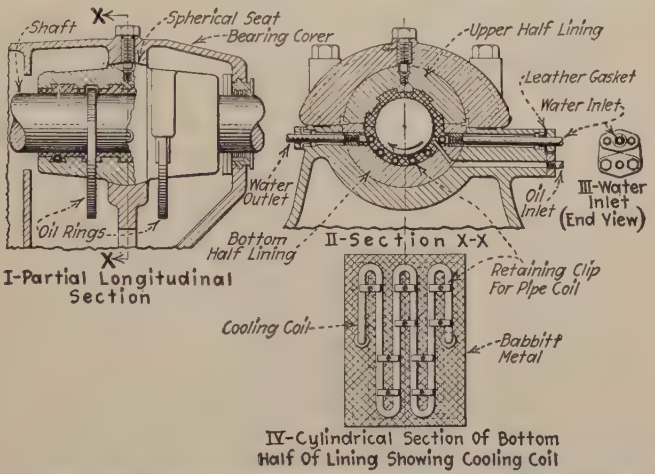


FIG. 101.—Showing constructional arrangement of a water-cooled steam-turbine bearing. (*General Electric Co.*)

coolers for lowering the temperature of the oil after it has passed through the bearing.

95. The Care Of The Main Bearings Of A Turbine consists principally in providing proper lubrication (see Div. 10). If proper lubrication is not maintained, excessive wear of the bearing will result, or the bearing may be burned out. Excessive wear in the bearing will disturb the alignment. This will usually cause undue vibration which will, in turn, cause the bearing to wear still more. If a slight misalignment due to wear is discovered in time it may be corrected by removing and inserting shims (Fig. 94) which are generally provided between the lining and the blocks which support the lining. With proper care, a turbine main bearing should last from 6 to 10 years.

Note.—Excessive Wear In The Bottom Of The Bearing Lining Usually Results In One Or More Of The Following Conditions: (1) *Misalignment of the bearings.* (2) *Shoulders along the oil groove,* which will cut off lubrication and cause heating. (3) *Contact between some stationary and some rotating part of the turbine.* (4) *Hard parts of the babbitt wearing the journal irregularly.* (5) *Excessive clearance in the upper half of the liner,* which may permit the oil to be thrown out. Obviously, the remedy is to install a new lining or to rebabbitt the old lining (Sec. 97).

96. A Turbine Bearing May Be Repaired by: (1) *Installing a new bearing which has been supplied by the manufacturer.* (2) *Rebabbitting the old bearing.* If a reserve bearing is at hand, the first method is the preferable one. This is because that, by using the reserve bearing, the necessary repair can be effected with a minimum loss of time, and also because a better fit will probably be secured. However, if an extra bearing is not readily available, new babbitt can be poured into the old lining and good results will obtain if the work is properly done. In any case, if a bearing is destroyed by any means except by ordinary wear, the cause of the destruction should be located and removed before the turbine is again put into service. A method of rebabbitting a turbine bearing is described in the following section. See also the author's Steam-engine Principles and Practice.

97. In Rebabbitting A Turbine Main Bearing, the original dimensions and shape of the old bearing should, if known (Fig. 97), be followed as closely as possible. If the original dimensions of the old bearing are unknown the new bearing can be made as explained below.

Explanation.—Pour the babbitt so that the diameter of the bearing is the same as that of the journal. Then scrape out the oil grooves. The oil grooves should (Fig. 102) be about $\frac{1}{8}$ in. deep and about $\frac{3}{8}$ in. wide. Two straight grooves (Fig. 102) are all that are generally required. Some turbine bearings have only one oil groove (Fig. 95) which is located in the top of the bearing. The location of the grooves will, for a forced-circulation lubricating system, be determined by the holes in the cast-iron shell for the oil inlet and outlet (Figs. 94, 96, 97, and 102). To prevent excessive oil leakage from the bearing, the ends of the groove should be about $\frac{1}{2}$ in. (Fig. 97) from the ends of the bearing. The square edges of the groove should be scraped away to a rounding contour (Fig. 102) so that there will be no sharp edge to interfere with the oil film.

After making the grooves, the next step is to fit the bearing to the journal. First put the lower half of the bearing in place. Then scrape out this lower half (see the author's STEAM-ENGINE PRINCIPLES AND PRACTICE) so that for about 55 deg. (Fig. 102) from each side of the vertical center line the bearing is an exact fit for the journal. Be sure to remove all high spots from this portion of the bearing. From the extremities of this area—that area which is fitted to the journal—up to the lower edges of the grooves, the bearing should be scraped away slightly (*A* and *A*, Fig. 102) so that a wedge-shaped oil-film space will be provided. A clearance should be provided between the journal and the upper half of the bearing. This clearance should be about 0.002 in. for each inch of journal diameter. That is, for a 2-in. journal, the clearance should be about: $2 \times 0.002 = 0.004$ in. This clearance may be

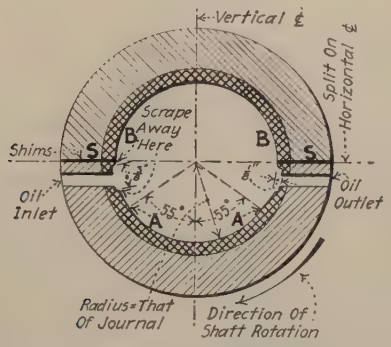


FIG. 102.—Illustrating one method of re-babbitting the main bearing of a turbine.

obtained by inserting shims (*S* and *S*, Fig. 102) of the proper thickness between the upper and lower halves of the lining, and then scraping away the bearing at *B* and *B*. If this clearance is too small the oil passage will be restricted. If it is too great, there may be an excessive oil leakage. The clearance above the journal can be determined by putting a piece of soft lead fuse wire on the top of the journal and then tightly bolting on the upper half of the lining. Then remove the upper half of the lining and caliper the mashed fuse wire.

98. The Primary Function Of Steam-turbine Thrust Bearings is to hold the shaft in such an axial position that proper clearance will be maintained between the rotating and stationary parts. Since impulse turbines are inherently subjected to but little end thrust and reaction turbines are generally provided with dummy pistons (Sec. 67) for balancing the end trust, the thrust bearings are not usually (see

Sec. 99) required to withstand much pressure. However, where a governor or an oil or water circulating pump is driven from the turbine shaft through a helical gear, a considerable end thrust may be exerted. Thrust bearings of the principal types, also some methods of adjusting them, are described in following sections.

99. The Four Principal Types Of Thrust Bearings which are used in steam turbines by the various manufacturers are: (1)

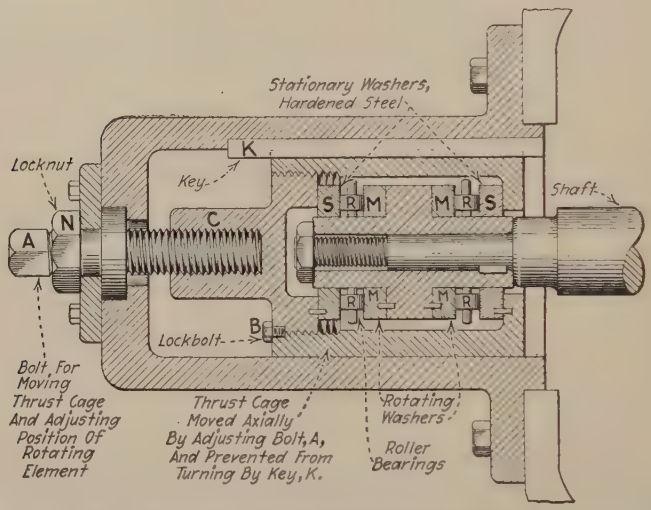


Fig. 103.—Roller thrust bearing. (The hardened steel washers, *S*, are held stationary by the dowels. The washers, *M*, rotate with the shaft. The rollers, *R*, roll between *M* and *S*. Clearance between *R*, *S* and *M* is adjusted by removing *B* and turning *C*. Axial position of shaft is adjusted by *A*.)

Roller thrust bearing, Fig. 103. (2) *Collar thrust bearing*, which may consist of only one collar, Fig. 104, or of a number of collars, Fig. 105. (3) *Ball thrust bearing*, Fig. 106. (4) *Kingsbury thrust bearing*, Figs. 107 and 108. The operation of the bearings of the first three types will be evident from a study of the respective illustrations. The operation of the Kingsbury thrust bearing is explained below.

Explanation.—The Kingsbury Thrust Bearing is sometimes (Fig. 108 and Fig. 69) contained within the main bearing lining. Sometimes it is mounted in a separate casing on the end of the shaft, as in Fig. 107.

This bearing is arranged to withstand thrust in the direction of arrow *A* against the bearing blocks, *F* (Fig. 108-*I*). One block, *G*, is placed on

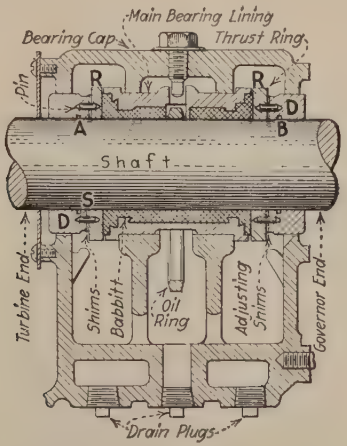


FIG. 104.—Simple collar thrust bearing. The two thrust rings *R* and *R* are pinned to the oil deflectors, *D*, and rotate with the shaft. Axial movement of the shaft is restrained by contact of these rings with the ends of the lining of the main bearing which are faced with babbitt. (*General Electric Co.*)

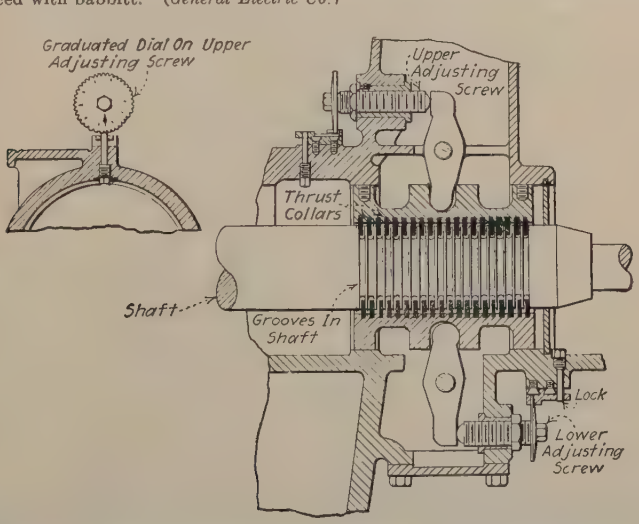


FIG. 105.—Multi-collar or marine-type thrust bearing. (*Westinghouse Electric & Mfg. Co.*)

the side opposite from the direction of thrust to restrain any endwise movement of the shaft. The bearing blocks, *F*, (as shown in *III*), rest

against pivoted projections on the equalizing blocks, *K*. The purpose of the equalizing blocks, *K*, is to equalize the pressure of each bearing

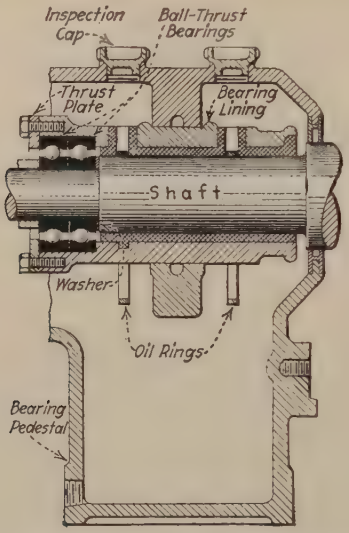


Fig. 106.—Showing the use of the Gurney Ball Thrust Bearing for maintaining axial shaft alignment. (*Terry* condensing turbine.)

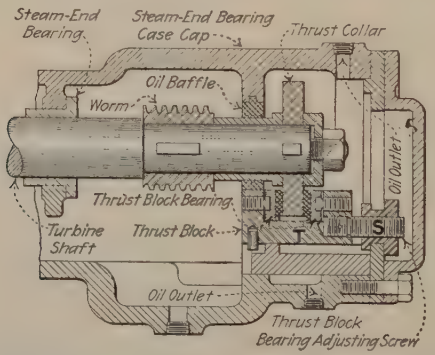


Fig. 107.—Kingsbury thrust bearing parts of *Moore* steam turbine (Instruction Card No. 3). Axial adjustment of the rotor is made by turning screw *S*, which moves a slide *T*, carrying the thrust bearing with it. The thrust blocks of this Kingsbury thrust bearing are also adjustable so the correct clearance between the blocks and the collar can be obtained. A clearance of about 0.004 to 0.005 in. on each side or 0.008 in. to 0.010 in. total, is recommended.

block on the collar, *E*. The collar, *E*, is shrunk on the shaft and rotates with it.

Thus if any portion of E (Fig. 108-*III*) tends to exert a greater pressure on some one bearing block, say block F, the equalizing block, K, on which this particular bearing block is pivoted is pressed downward. This causes the two adjacent equalizing blocks K_1 and K_1 to rotate a little, which causes the next equalizing blocks K_2 to push upward on the next bearing block, F_2. Thus, the total thrust which is exerted by E in the direction of A (Fig. 108) is always equally divided between all of the bearing blocks. Also, the total thrust on any one bearing block is uniformly distributed over the face of that block. Consequently every

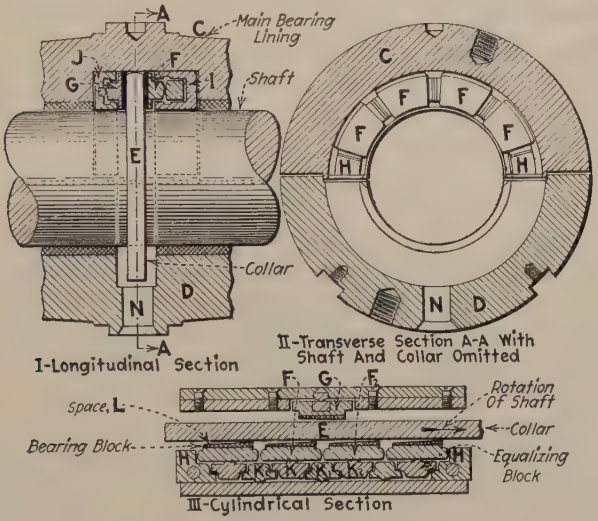


FIG. 108.—Kingsbury thrust bearing.

minute portion of the face of each bearing block is always active in carrying the thrust. This design and construction produces (automatically) a wedge-shaped oil film at L (Fig. 108-*III*), which provides effective lubrication at all times.

Inasmuch as a thrust bearing of this type is capable of satisfactory operation under very high unit pressures (350 to 500 lb. per sq. in.), the area of the balance pistons of reaction turbines is sometimes reduced and the Kingsbury thrust bearing is designed to carry the unbalanced end thrust. To insure that the end thrust will always be against the bearing blocks, turbines are (*Westinghouse Electric & Mfg. Co.*, "INSTRUCTION BOOK No. 5,171"), sometimes installed with the thrust-bearing end lower than the other end by about 0.02 in. per foot of length of the turbine.

100. The Axial Adjustment Of A Turbine Rotor Determines The Axial Clearance Between The Rotating And The Sta-

tionary Parts Of The Turbine Proper. If the axial position of the rotor is not properly adjusted, the turbine will not operate at its maximum efficiency; and in case of extreme axial mal-adjustment, the turbine may be wrecked. As suggested in Sec. 98, the axial adjustment of the rotor is generally made by shifting the thrust bearing. Various methods of making this adjustment and the amount of clearance which is necessary between stationary and moving parts of the turbine are discussed in following sections.

101. The Types Of Mechanisms Which Are Generally Employed For Axial Adjustment Of A Turbine Rotor are: (1) *Screws or nuts*, Figs. 103, 105, and 107. (2) *Shims*, Fig. 104;

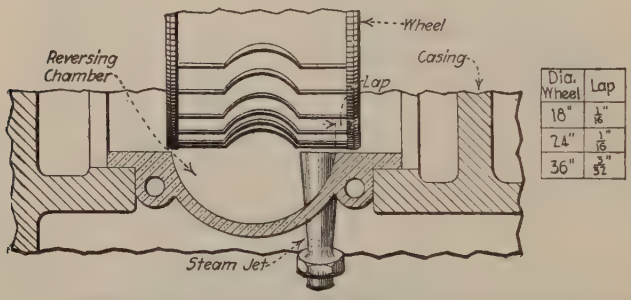


Fig. 109.—Showing correct amount of "lap" for a *Terry* tangential-flow turbine.

see also Fig. 57. Screws are used for effecting axial adjustment in practically all steam turbines. However, in turbines of the smaller capacities, some manufacturers use shims (Fig. 104). The detailed design of the mechanism for axial adjustment which is used by one manufacturer differs from that used by every other manufacturer. Even the axial-adjustment mechanisms for turbines made by the same manufacturer, but of different types and capacities, are different one from the other. Consequently, it is impractical to treat herein the various mechanisms which are employed for this purpose. The operator should, by a careful study of the machine and of the manufacturer's instructions, thoroughly familiarize himself with the adjusting mechanism before attempting to make an axial adjustment of the rotor.

102. The Axial Adjustment Of A Tangential-flow Turbine (Sec. 56) is usually made by providing the proper "lap" between the wheel and the reversing chamber. See Figs. 60 and 109.

103. The Axial Adjustment Of An Axial-flow Turbine Rotor Which Is Provided With Adjusting Screws is usually made as follows: The turbine after being heated to its operating temperature (Div. 11) is throttled down so that it runs at about 10 per cent. of its normal speed. While running at this decreased speed, the rotor is, by the axial adjusting mechanism, moved in a longitudinal direction until a slight rubbing is heard. Then, the adjusting mechanism is operated in the opposite direction until a slight rubbing is again heard. In making this second movement count the number of nut or screw turns which are made. Now, move the rotor back in the direction of the first movement by one-half the number of nut or screw turns just counted. Next, by whatever kind of locking device that is provided, lock the rotor in this position. This should locate the moving part in the center of its minimum clearance, which, for most axial-flow turbines, is the correct axial position for the rotor.

Note.—The Slightest Rubbing May Be Readily Heard by holding one end of a short piece of gas pipe or a file against the casing and the other end near the adjuster's ear. This rubbing should not be permitted to continue longer than an instant, and should not be severe. Otherwise, the turbine is likely to be damaged.

Example.—Assume that the rotor of the 3,600-r.p.m. turbine the thrust bearing of which is shown in Fig. 103 is to be axially adjusted. First heat up the turbine. Then, throttle down to about 350 or 400 r.p.m. Turn bolt *A* in a right-hand direction until a rubbing is heard. Then, counting the number of turns, turn *A* back in a left-hand direction until a rubbing is again heard. Now turn *A* in the right-hand direction one-half the number of turns just counted. Next, lock *A* in this position with the locknut, *N*.

Note.—The Axial Rotor-adjustment Of Those Turbines Which Have Axial-clearance Metallic Labyrinth Glands (Sec. 112) Cannot Be Made As Described Above. This is because the axial-clearance labyrinth glands (Fig. 112) must have a small axial clearance between the rings, *R*, on the balance piston and the tips, *T*, of the dummy rings. The proper value of this clearance varies with the size and design of the turbine and must be obtained from the turbine manufacturer.

104. The Axial Adjustment Of A Turbine Rotor Which Is Provided With Adjusting Shims (Fig. 104) must be made in a manner somewhat different from that which is described in Sec. 103. This is because the shims are not readily accessible, and therefore the adjustment cannot generally be made while the turbine is at its operating temperature. Consequently, the adjustment must be made while the turbine is relatively cold. Therefore, in effecting the adjustment while the turbine is cold, due allowance must be made for the expansion of the shaft and casing which will occur when the turbine is heated to its operating temperature. Usually this allowance may be made by adjusting the axial position of the rotor as explained below.

Explanation.—Take down the bearing (Fig. 104) and fill the shim-spaces at *A* and *B* with shims so that one-third of the thickness of all shims will be in the space at *B* and two thirds of the thickness of all shims will be in the space at *A*. This will (Fig. 70) locate the rotating discs to the right of the central position. Then, when the turbine is heated during operation, the expansion of the shaft occurs away from the thrust bearing toward the exhaust end. This expansion moves the discs to the left (Fig. 70), and the rotating blades will take a position nearly central with the stationary blades and nozzles.

105. Thrust Bearings Of The Collar, Roller, And Kingsbury Types Must Be Adjusted after the axial adjustment of the rotor is made. If the thrust bearing is too tight, it will bind and may burn out. If it is too loose, the correct axial position of the rotor will not be maintained. A thrust bearing of the collar, roller, or Kingsbury type should have a total clearance of from about 0.008 to 0.010 in. That is, the thrust bearing should be so adjusted that the shaft will have a "play" in the axial direction of from about 0.008 to 0.010 in. This adjustment may be made by screws which are (Fig. 107) provided for this purpose; or in the case of a shimmed bearing (Fig. 104) which has no screws, a 0.004-in. shim is, after the axial adjustment (Sec. 104) has been made, removed from each of the shim-spaces.

Note.—Ball Thrust Bearings Usually Require No Adjustment. They are made with the proper amount of clearance. When they wear so that the clearance is excessive, they must be renewed; see Sec. 93.

106. A Steam-turbine Gland is a device for minimizing the leakage of steam or air through the clearance which must be provided between the rotating and stationary parts. Thus, where the shaft passes through the high-pressure end of the casing (Fig. 75) a gland must be provided to prevent an excessive leakage of steam out of the turbine. In a pressure-staged or a velocity-and-pressure-staged turbine (Fig. 70) the pressure on one side of a diaphragm is less than the pressure on the other side. Consequently, there is a tendency for the steam to leak past the diaphragm along the periphery of the shaft. To minimize the steam leakage at these locations, a gland of some sort must be used. When a turbine is operated condensing, the steam pressure within the turbine casing at the exhaust end is less than atmospheric pressure. Therefore, to prevent air from leaking into the turbine and decreasing the vacuum, a gland must be provided around the shaft where it passes through the exhaust end of the turbine. The repair and adjustment of glands of various types are described in the following sections.

107. There Are Four Principal Types Of Steam-turbine Glands: (1) *Metallic-packed or stuffing-box gland*, Fig. 110. (2) *The metallic-labyrinth gland*, Fig. 113. (3) *The centrifugal water-packed gland*, Fig. 116. (4) *The carbon-packed gland* (Fig. 120). The construction and maintenance of glands of each of these types are treated hereinafter in this division.

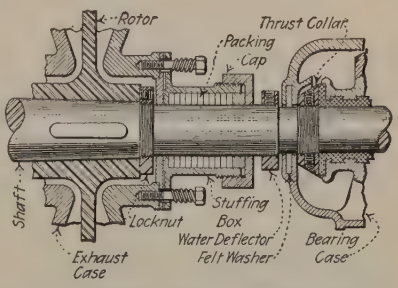


FIG. 110.—Section through stuffing box and related parts of Type-6 *Sturtevant* turbine.

108. Metallic-packed Or Stuffing-box Glands (Fig. 110) are stuffing boxes which are packed with a flexible metallic packing. Glands of this type are, generally, used only for velocity- or single-staged turbines which are designed to operate non-condensing at low back-pressures—not exceeding about 10 lb. per sq. in.—and at speeds below 3,600 r.p.m. Since the steam pressure in the casing of a turbine of this type

is about the same as the atmospheric pressure, the function of the glands is not so much to prevent a waste of steam as it is to prevent any steam which condenses on the shaft from ultimately finding its way into the bearing.

109. A Metallic-packed Gland May Be Repacked as described below. Soft metallic packing rings with skived joints (Fig. 111) should be used. The rings should be about ¼ in. thick. The outer diameter of the rings should be approximately the same as the inner diameter of the stuffing box. The inner diameter of the rings should be approximately the same as the diameter of the shaft. The number of rings

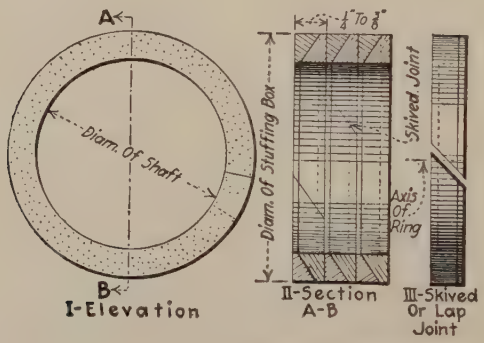


Fig. 111.—Showing skived-jointed metal packing rings.

required will depend upon the length of the stuffing box and upon the thickness of the rings which are used. After new packing is installed, the cap (Fig. 110) should be screwed up as tightly as possible with the fingers. Then, after the turbine has been started, the cap may be tightened a little more with a wrench. Allow a reasonable time for the packing to adjust itself before making any further adjustments. All packing of this type will leak somewhat when the turbine is starting cold but the packing becomes tighter as the turbine heats. If the cap is screwed up too tightly, the packing will be scorched and ruined. Never use a wrench to tighten the gland except when the turbine is running. Unless a packing should burn out, it is seldom necessary to install an entire new packing; merely add a new ring as described below.

NOTE.—THE WEAR IN A METALLIC-PACKED GLAND SHOULD BE TAKEN UP by tightening the stuffing-box cap and occasionally inserting a new ring. When a new ring is inserted, it should be placed between the outer and the second rings of the old packing. A slight steam leakage from a metallic packed gland is permissible and helps to lubricate the gland. But a leak that "blows" steam should not be tolerated.

110. Metallic-labyrinth Glands (Figs.112, 113 and 114) are, as the name suggests, designed to force the steam to follow a

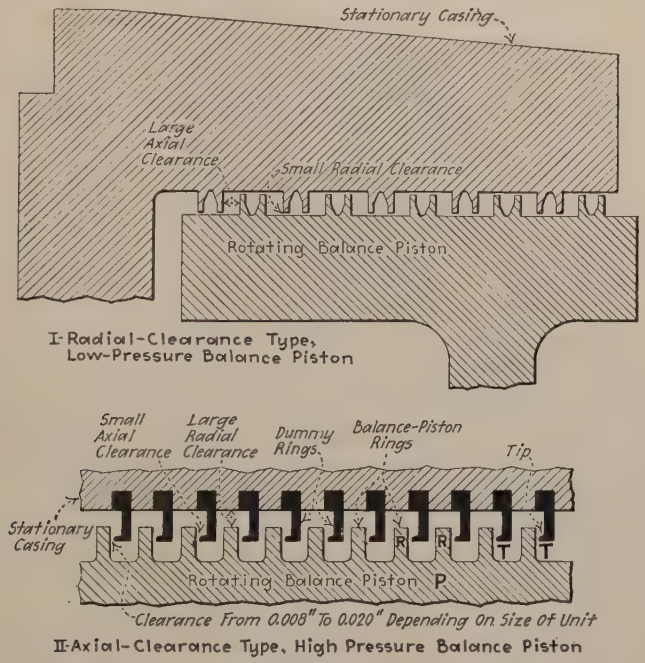


FIG. 112.—Double-labyrinth glands to minimize steam leakage around the balance pistons in a reaction turbine. (*Allis-Chalmers Mfg. Co.*)

long winding path through the gland. The steam, in passing through each constriction in the path, is subjected to a throttling action with a consequent reduction in pressure. Thus, the reduction in pressure and the frictional resistance which are occasioned in passing through the labyrinth passageway permit but a small amount of steam to escape.

111. There Are Two Types Of Metallic-labyrinth Glands: (1) *The double labyrinth gland*, Figs. 112 and 113, which consists of annular rings on the rotating element which fit into annular grooves in the stationary element. (2) *The single labyrinth gland*, Fig. 114, which consists of a number of stationary annular saw-toothed projections which fit closely

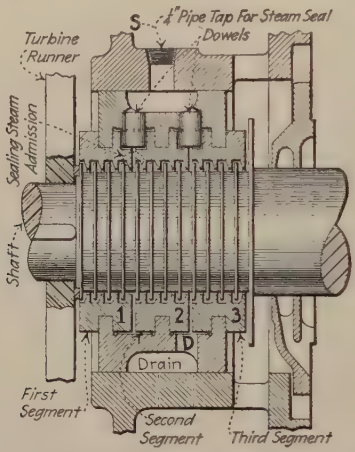


Fig. 113.—Gland of *Steam Motors Company* turbine. This gland is of the double-labyrinth type and is used in all of this company's turbines. It is suitable for any ordinary back pressure or vacuum. A drain, *D*, is provided between segments 2 and 3 of the packing. This should be piped and the drain line led away to the atmosphere. Or, since only clean steam drains from it, it may be led to the feed-water tank, provided the tank is operated at atmospheric pressure. For location of drain, see Fig. 28. For condensing operation a ¼-in. steam-seal pipe, in which is inserted a valve, should be led to *S*. The sealing steam is admitted to the gland between segments 1 and 2 and the valve should be opened sufficiently wide, that there is just a "whiff" of steam leakage visible at the bearing end of the gland. The drain connection remains in any case, but for condensing operation it may be fitted with a valve which should be so adjusted that the flow of steam through the drain will not be excessive.

to the smooth shaft. Glands of each of these types are described in the following sections.

112. The Double Labyrinth Glands (Figs. 112 and 113) are generally used: (1) *To prevent leakage of steam over the balance pistons* (Sec. 67 and Fig. 112) *in a reaction turbine;* as indicated in Fig. 112, balance-piston labyrinth glands may be of either the *axial-clearance* or of the *radial-clearance* type. (2) *To prevent steam or air leakage around the shaft* (Figs. 65

and 113) at the steam- or exhaust-end of an impulse turbine. If the turbine is to be operated non-condensing against an appreciable back pressure, steam leakage in minimized by the labyrinth passageway as explained in Sec. 110. If the turbine is to be operated condensing, leakage of air into the casing (along the shaft) is prevented by a steam seal, the operation of which is explained below.

EXPLANATION.—THE OPERATION OF A STEAM SEAL is as follows: Assume that steam is admitted at about the middle of the gland (*S*, Fig. 113) at a pressure of 3 lb. per sq. in. gage. The steam will leak through the labyrinth passageway in both directions, part of it going into the turbine and part outward to the atmosphere. If steam is leaking outward to the atmosphere, it is obvious that air cannot at the same time leak into the turbine casing. The steam which leaks into the casing will have practically no effect on the vacuum, whereas air would, if permitted to leak in, tend to lower the vacuum considerably. The operation of the steam seal in a carbon-packed gland (Sec. 118) is essentially the same as is described above.

NOTE.—THE ADVANTAGES AND DISADVANTAGES OF A DOUBLE LABYRINTH GLAND are: (1) *There are no rubbing surfaces.* Therefore it is frictionless and consequently has a long life. (2) *It ordinarily limits the axial end-play of the shaft.* Hence, if rubbing should occur and the gland is injured, a new gland will usually be required. The installation of a new gland is an extremely difficult and expensive procedure.

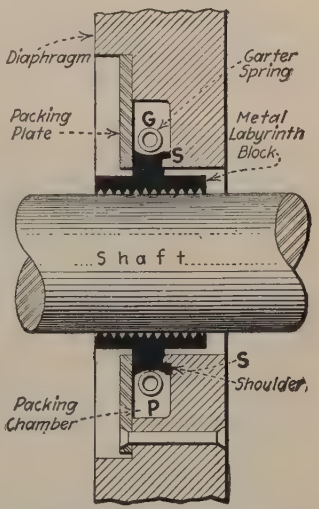


FIG. 114.—Single-labyrinth-type packing gland to prevent steam leakage along the shaft where it passes through a diaphragm.

113. The Single Labyrinth Packing Gland (Fig. 114 and Sec. 111) consists of one or more metallic rings (Fig. 115) which are loosely supported by a shoulder (*S*, Fig. 114) in the packing chamber. Each ring is composed of three equal segments (*X*, *Y*, and *Z*, Fig. 115) which are held together by a garter spring (*G*, Fig. 114). One of the segments is provided with a stop to prevent the ring from rotating with the shaft. When first

assembled, each ring is so machined that the tips of the saw-tooth projections hug the shaft, and the flange (*F*, Fig. 115) clears the shoulder, *S*, in the packing chamber (*P*, Fig. 114). When cold, the clearance between the segments of the ring is (Fig. 115) about 0.005 in. When the turbine heats under operating conditions, the rings expand. Thus, the clearance between segments closes up and forms an *arch bound butt-joint.* Also, the shaft wears off the points of the teeth until the flange (*F*, Fig. 115) on the ring rests on the rabbeted shoulder, *S*, in the packing chamber, *P*, Fig. 114. Thus, a closely fitting labyrinth gland is provided, the applications of which are given below.

Note.—Single Labyrinth Packing Glands Are Used In Some Impulse Turbines Of The Smaller Capacities for both the steam and

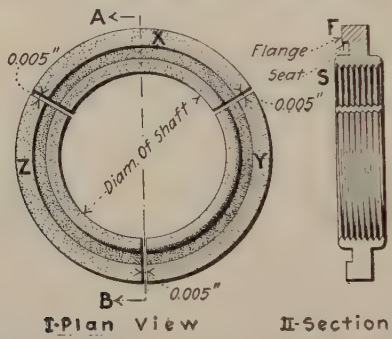


Fig. 115.—Metal packing ring of the single-labyrinth type.

exhaust-end glands and also for the diaphragm glands. When used in a turbine that is to be operated condensing, single labyrinth glands for the steam and the exhaust-ends are steam sealed in a manner which is similar to the steam seal used for double labyrinth glands (Sec. 112).

Note.—Single Labyrinth Glands May Be Tightened To Take Up Wear by machining out the flange seat (*S*, Fig. 115) of the ring, and then filing off the ends of the segments so that the correct end clearance of about 0.005 in. (Fig. 115) between segments will be provided. These operations should be performed with extreme care so that concentricity and proper end-clearance will be maintained. If the ends of the saw-teeth are worn so that the tips are materially widened, the grooves between teeth should be remachined out so that the teeth are sharp.

114. A Centrifugal Water-packed Gland (Fig. 116) is merely a centrifugal-pump runner, *C*, which is fixed to and rotates with

the turbine shaft. Machined in the turbine casing, or in the gland casing, is a chamber, B, within which the runner rotates. Water is admitted at the inlet, A. The runner is so designed that when the turbine is operating at normal speed, a water pressure of about 20 lb. per sq. in. gage would, if the water

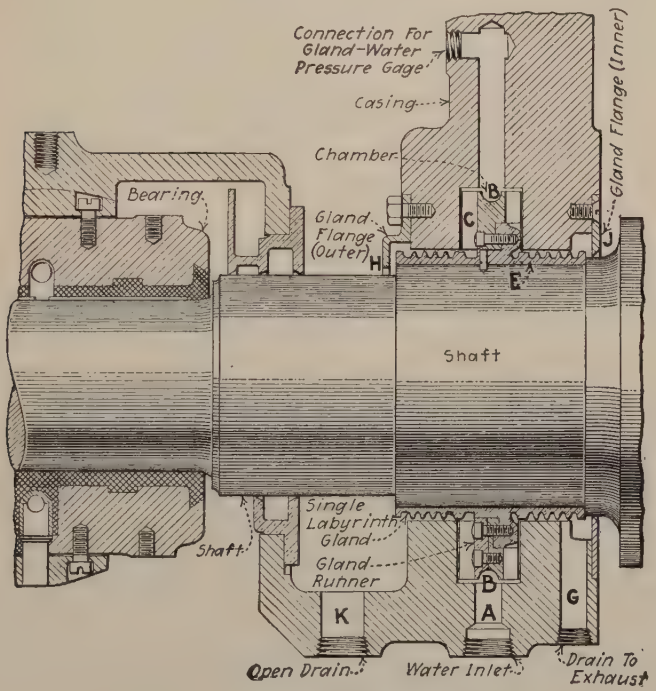


Fig. 116.—Centrifugal water-packed gland.

were admitted at the center of the runner and no outlet were provided, be produced at the periphery of B. Consequently, if water is supplied at the periphery at a pressure of about 5 lb. per sq. in. gage, the pump runner holds the water in a solid annular ring against the periphery of the chamber, C. This produces a hermetic seal which entirely precludes leakage.

Note.—Any Water Leakage From A Centrifugal Water-packed Gland Must Be Drained Away. If the turbine is to be operated condensing, the glands must sometimes be sealed for raising the vacuum before the turbine is started. Obviously, during the period of starting

the pump runner does not function. Therefore, to prevent excessive leakage of the sealing water while starting, single labyrinth glands are provided as shown at *E* in Fig. 116. There may also be a slight leakage of water while the turbine is running at full speed. To prevent any water which may leak outward (to the left in Fig. 116) along the shaft from being thrown out into the engine room, an outer gland flange, *H*, is provided. To prevent water from finding its way along the shaft and into the bearing, the drain *K* is piped to an open sewer or to some other region where the pressure is not above atmospheric. This drain, *K*, must be kept open at all times. The inner flange (*J*, Fig. 116) prevents any water which leaks inward from being thrown against the moving blades. The drain, *G*, is piped to the exhaust pipe of the turbine. Other methods than those described above for sealing during starting and for taking care of the leakage water, will be evident from the construction of the turbine in which they are used.

115. Centrifugal Water-packed Glands Cannot Be Used In Close Proximity To High-pressure Steam. That is, if a centrifugal water-packed gland were used in the high-pressure end of a pressure-staged or composite-staged turbine (Fig. 73), the water in the gland would tend to vaporize. Consequently, glands of this type are generally used for only: (1) *The exhaust-end of impulse turbines*, Fig. 69. (2) *The steam- and exhaust-ends of low-pressure impulse turbines*, Sec. 35. (3) *Both ends of reaction turbines.* The water in a centrifugal water-packed gland must, when used in close proximity to steam which is above atmospheric pressure, be circulated and cooled to keep it from vaporizing. For a gland of this type that is used on the exhaust end of a turbine which is operating condensing, the water does not need to be circulated.

116. The Gland Sealing Water For A Centrifugal Water-packed Gland Must Not Contain Any Sediment Or Scale-forming Salts. This is because if the water does contain such substances, the centrifugal action and the heat will cause the solids to be deposited in the gland in the form of scale. The scale will clog the gland and frequent disassembling and cleaning will be required. If scale is formed within the gland chamber and allowed to accumulate, the runner will eventually rub and cause excessive vibration and leakage; or in extreme cases, the runner may be broken.

117. The Arrangement Of The Gland-water Piping (Fig. 117) will depend upon the available supply of pure soft water.

However, the general scheme which is usually employed, consists of a tank or reservoir, R, located at a sufficient height above the glands so that the proper water pressure in the glands will be provided by gravity. One such arrangement is shown in Fig. 117. Where the only available supply of pure water is that for boiler feeding, and the condensed steam is pumped directly back to the boiler, the gland-water reservoir may be supplied from the delivery of the condensate pump. In such cases, the gland-water reservoir should be of sufficient

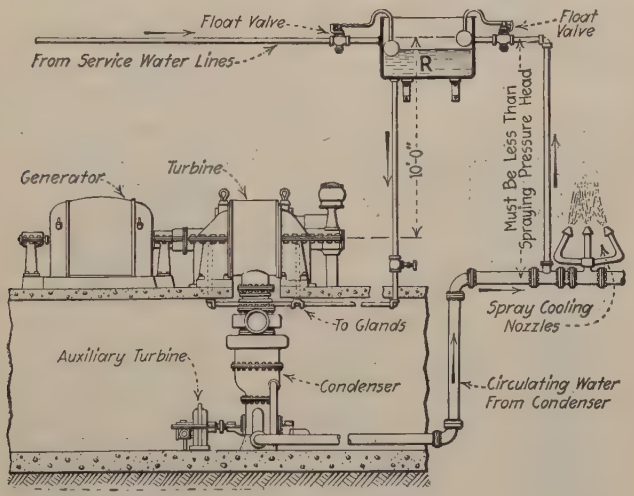
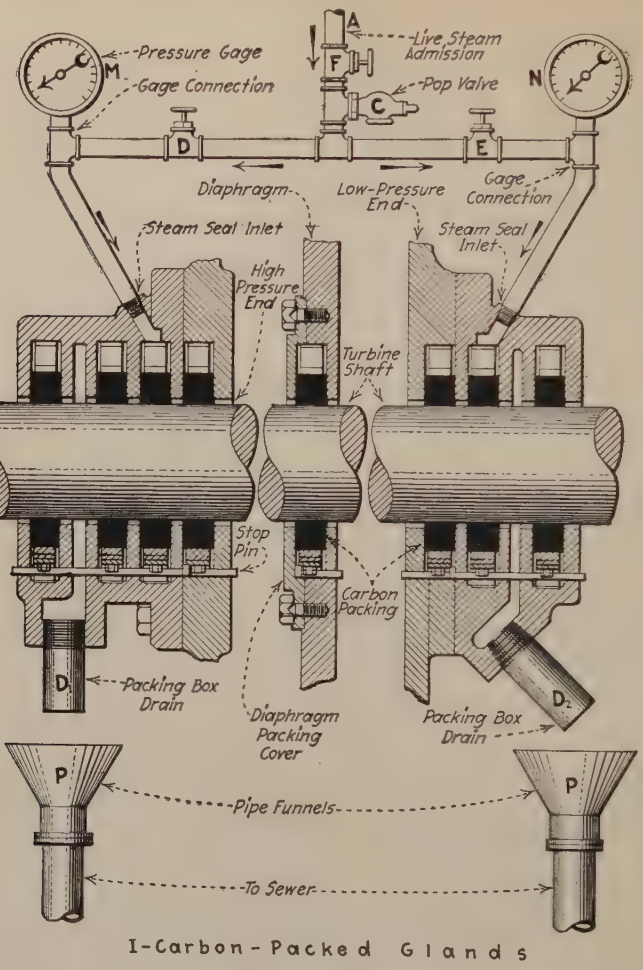


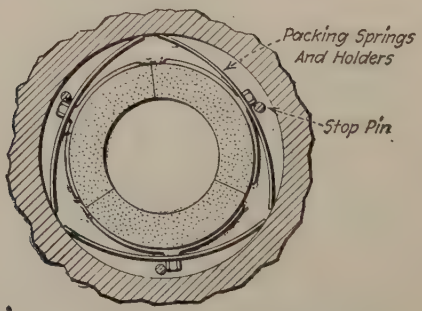
Fig. 117.—Piping arrangement for centrifugal water-packed glands.

capacity so that the water which is delivered to it will have ample time to cool before it enters the glands. Where the water must flow through the gland (Sec. 115), the discharge may be piped to a feed-water tank or to the hot-well.

118. Carbon-packed Glands (Fig. 118) may be used for packing the steam-end and the exhaust-end of turbines of all types, and also for packing the diaphragms of pressure-staged or of velocity-and-pressure-staged turbines. Carbon-packed glands which are used in the steam- and exhaust-ends of condensing turbines are generally provided with a steam seal (Sec. 112). The steam which leaks through the glands and condenses must be drained away. Steam-seal piping, drain-



I-Carbon-Packed Glands



II-Section Through Carbon Packing Ring

FIG. 118.—Carbon-packed glands in head-end, exhaust-end, and diaphragms of a pressure-and-velocity staged impulse turbine. The steam-seal piping is also shown. *General Electric Co.*)

age piping, operation, and repair of carbon-packed glands are treated in the following sections of this division.

119. The Construction Of Carbon-packed Glands varies according to the conditions under which they are to be used, and also according to the manufacturer. Carbon-packed glands (Fig. 119) consists of one or more carbon rings which are contained in a chamber, *C*. The carbon rings encircle and fit closely to the shaft, *S*. They are made, usually, in three equal segments (Fig. 118-*II*) which are butt-jointed one to the other. These segments are held together either by a garter spring (*B*, Fig. 120) which completely en-

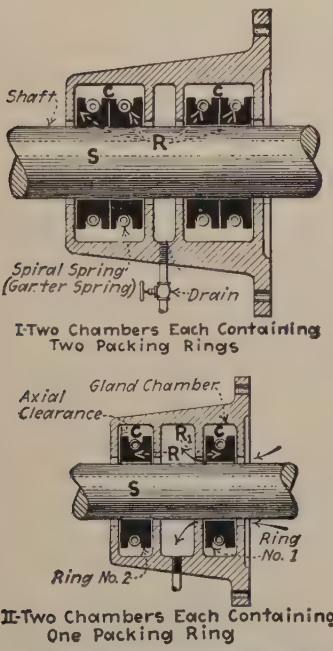


FIG. 119.—Showing various arrangements of carbon rings in carbon-packed glands.

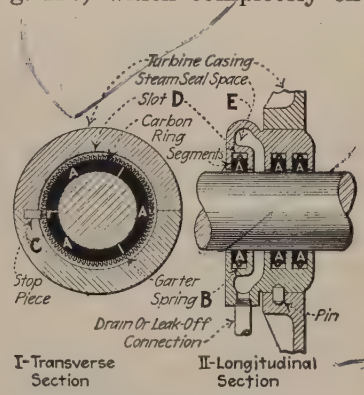


FIG. 120.—Carbon-ring glands of the *Terry* turbine.

circles the ring, or by three flat tangential springs (Fig. 118-*II*) which bear against the inner periphery of the chamber. The chamber is provided with one or more lugs *C*, Fig. 120, or straps which engage with a lug or keyway that is carried by the ring, thus preventing the ring from rotating with the shaft. Carbon-packed glands which are used in the diaphragm of a pressure-staged turbine generally consist of only one ring. But the head- and exhaust-end glands may comprise any one of various arrangements, (Fig. 119) such as two

chambers containing one ring each, two chambers containing two rings each, three chambers, containing one ring each (Fig. 120) etc.

120. The Steam-seal Piping Of Carbon-packed Glands may be arranged as indicated in Fig. 118. The live-steam admission, A, is taken from the boiler side of the main throttle valve. The pop valve, C, is set to blow at a pressure of about 10 lb. per sq. in. gage. If the turbine is to be operated non-condensing, the steam seal is not required, consequently the globe valve, F, may be closed. If the turbine is to be operated condensing, and the vacuum is established before the turbine is started, then both the head- and exhaust-end glands should be steam sealed. To effect this seal, valves F, D and E are opened so that the gages at M and N read about 3 lb. per sq. in., or so that a slight steam cloud issues from both packing-box-drain pipes, D_1 and D_2. Then when the turbine is brought up to speed, valve D may be closed. The packing-box drains, P and P, should be piped to a region wherein the pressure will never be above that of the atmosphere. Although there are other arrangements of steam-seal piping, they will not be treated in this book.

Note.—The Steam Leakage At The Drains Of Steam-sealed Glands Should Preferably Be Visible From The Turbine Room as suggested at P, Fig. 118. Such an arrangement will enable the attendant to readily observe the amount of steam which is issuing from the glands. It is desirable that there be a slight leakage of steam (just a trace of visible water vapor) from carbon-packed glands. This provides a sort of lubrication for the carbon rings. Also, unless some steam is leaking from the exhaust-end gland of a condensing turbine, air is probably leaking into the turbine. If the steam leakage from the exhaust-end gland is excessive when the gland-pressure gage reads about 3 lb. per sq. in., the carbon rings should be refitted. If an excessive amount of steam leaks from the head-end gland, these rings should be refitted. About the only way to determine whether or not diaphragm carbon packing (Fig. 118) needs refitting is, when the turbine casing is opened for inspection, to check the clearance (Sec. 121) with a thickness gage. Methods of refitting carbon packing rings are discussed in the following sections.

121. The Diametral Clearance Between A Carbon Ring And The Shaft should be about 0.002 in. per in. of shaft

diameter when the shaft is cold. This will, due to shaft expansion, provide a total diametral clearance of approximately 0.000,5 to 0.001 in. when the turbine heats up during operation. For high pressures and superheat, the diametral clearance should be about 0.003 in. per in. of the cold-shaft diameter. On small capacity turbines—up to about 100 kw.— the rings may be bored to approximately the cold-shaft diameter. Then, after two or three hours run, they will wear to normal size and an extremely accurate fit will result. However, this procedure is not advisable for large turbines because, if the rings pinch the shaft of a large turbine, serious heating and vibration may be caused.

Note.—The Axial Clearance Of Carbon Packing Rings (Fig. 119) should be from about 0.003 to 0.006 in. That is, the width of the groove in the packing casing, as measured in an axial direction, should exceed the axial thickness of the carbon ring by this amount. If the clearance is too small, rust and sediment are likely to cause the ring to stick. If the clearance is too large, the steam pressure may not hold the ring tightly against the side of the groove, and steam will leak around the outside of the ring.

122. A Mandrel Will Be Found Extremely Convenient In Fitting A Carbon Packing Ring.—The diameter of the mandrel should be the exact size to which the ring is to be fitted. The correct diameter may be determined by the amount of the required clearance as stated in Sec. 121. A piece of iron pipe can easily be turned to the proper diameter. The ring can then be easily and accurately fitted around this mandrel.

123. In Refitting A Carbon Packing Ring which has worn too large, the inner diameter must be decreased. This may be done by filing off the joints (Fig. 121) and then reboring, as hereinafter explained so that the inner periphery of the ring will be a true circle of the proper diameter. When the rings are but slightly worn so that the diameter does not have to be decreased more than about 0.004 or 0.005 in. it is not necessary to rebore. The joint surfaces at the ends of the segments may be filed off and the ring assembled on the shaft. Then the shaft will wear the inner surface of the ring to a true circle. For methods of decreasing the inner diameter, see note below.

NOTE.—IN FILING OFF THE SURFACES OF THE JOINTS OF A CARBON PACKING RING it is of paramount importance that the finished surfaces of each joint be true as shown in Fig. 122. To assist in filing a true surface, a wooden jig or pattern may be made and used as indicated in Fig.

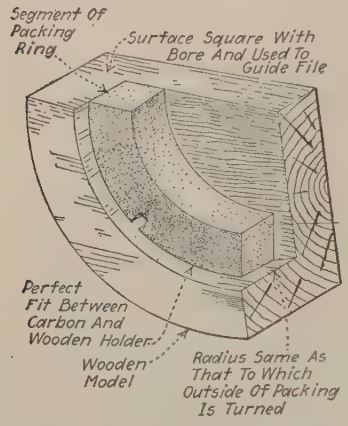

Segment Of Packing Ring
Surface Square With Bore And Used To Guide File
Perfect Fit Between Carbon And Wooden Holder
Wooden Model
Radius Same As That To Which Outside Of Packing Is Turned

FIG. 121.—Wooden jig for holding carbon packing rings for filing the joint surface. (E. H. Thompson in *Power*, Sept. 21, 1920.)

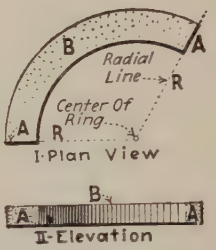


FIG. 122.—Joint surfaces of each segment of a carbon packing ring must be made true. (The plane of surfaces *A* should be perpendicular to the plane of surface *B*. Surfaces *A* should also coincide with a radial line *R*.)

121. If such a jig is not available, the three or four segments of a ring may be held in a vise as shown in Fig. 123 being careful to line up the joint surfaces of all of the segments. The relatively large area thus provided by the ends will assist materially in guiding the file. Be careful

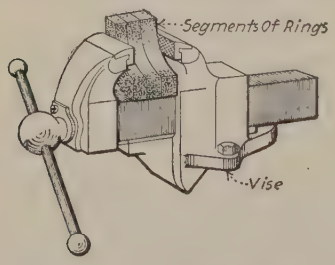


FIG. 123.—Carbon packing rings fitted in vise preparatory to filing the joint surfaces. (E. H. Thompson in *Power*, Sept. 21, 1920.)

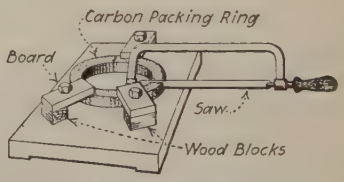


FIG. 124.—Hacksaw used to decrease the diameter of a carbon packing ring. (E. H. Thompson in *Power*, Sept. 21, 1920.)

not to screw the vise up too tightly as the rings are likely to be broken. If the vise jaws are rough, they may be lined with sandpaper. If the joint surfaces require a considerable amount of dressing down, the entire ring may be clamped on a board (Fig. 124). Then, with a hacksaw,

cut through each joint, keeping the saw in a radial and vertical position so that the blade lines up with a diameter of the ring.

Note.—The Carbon Ring Should, Usually, Be Bored Out To The Proper Diameter (Sec. 121) after the joint surfaces have been dressed down as explained above. The three segments of the ring are assembled and the boring done on a lathe. A large strong ring with its segments held together with the spiral spring or with a wire wound around its outer circumference may sometimes be held in the lathe chuck for reboring. But the best method is probably to make a wooden chuck by clamping a wooden block in the lathe chuck or in its faceplate, and then boring a cavity in the block, into which the ring will just fit. The bored surface of the ring should be made smooth by polishing it with No. 00 sandpaper. Emery cloth should not be used on the packing rings because particles of emery will stick to the ring and then cut the shaft. If by accident the ring is bored out a little too large, the joints may be dressed down as explained above, and no reboring will be required.

QUESTIONS ON DIVISION 5

1. Why does the satisfactory operation of a steam turbine depend largely upon the condition of the shaft, bearings, and glands?

2. How are the shafts of impulse turbines generally constructed? Of reaction turbines?

3. What is meant by the *critical speed* of a turbine shaft?

4. What is meant by a *flexible* turbine shaft? By a *stiff* shaft? Do most modern turbines have a flexible or a stiff shaft?

5. What are the two principal types of *steam-turbine bearings?*

6. Make a table showing the classification of steam-turbine bearings.

7. Make a sketch of and name the principal parts of a *plain, flexible, steam-turbine main bearing.*

8. Make a sketch to explain the operation of a "flexible" bearing.

9. In what kind of turbines are *ball bearings* sometimes used as main bearings? If a ball bearing becomes worn, what must be done?

10. In general, what is the maximum temperature at which a main bearing should be operated? Name two means which are used to reduce the temperature of turbine bearings.

11. What attention is necessary for the successful operation of a main bearing?

12. Name five things which are likely to result from excessive wear of a bearing lining.

13. Name two methods of repairing a turbine bearing.

14. Explain with a sketch how a turbine bearing may be rebabbitted.

15. What is the primary function of a steam turbine *thrust bearing?*

16. Name four principal types of thrust bearings.

17. Explain with a sketch the operation of the *Kingsbury* thrust bearing.

18. What determines the *axial clearance* between the rotating and the stationary parts of a steam turbine? What is likely to happen if proper clearance between the moving and stationary parts is not maintained?

19. What two types of mechanisms are generally employed for the axial adjustment of a turbine rotor?

20. How is the correct *axial adjustment* of a tangential-flow turbine generally determined?

21. Explain how the axial adjustment of an axial-flow turbine rotor which is provided with adjusting screws is usually made.

22. Explain how the axial adjustment of a turbine rotor is made with adjusting shims.

23. Why must the thrust bearing itself have some clearance?

24. How much clearance is usually allowed in a thrust bearing?

25. What is a *steam-turbine gland?*

26. What are the functions of a gland?

27. Name four principal types of turbine glands.

28. Make a sketch of a *metallic-packed* or stuffing-box gland.

29. For what types of turbines and under what operating conditions are metallic-packed glands used?

30. Explain how to repack a metallic-packed gland. Make a sketch of a skived joint.

31. Explain how the wear in a metallic-packed gland may be taken up.

32. Make a sketch of and explain the action of a *metallic labyrinth gland.*

33. What are the two principal types of metallic labyrinth glands?

34. For what purposes and in what kinds of turbines are double labyrinth glands used?

35. Explain the operation of a *steam seal.*

36. State the advantages and disadvantages of a double labyrinth gland.

37. Describe the packing ring used in a single labyrinth gland.

38. Explain how a single labyrinth gland may be refitted after it has become worn.

39. What is a *centrifugal water-packed gland?* Explain its operation.

40. Why cannot a centrifugal water-packed gland be used in close proximity to high-pressure steam? If a centrifugal water packed gland is to be used close to steam above atmospheric pressure, what means are employed to prevent the water in the gland from vaporizing?

41. What must be the condition of the gland sealing water?

42. Upon what will the arrangement of the gland-water piping depend?

43. Where may *carbon-packed glands* be used?

44. Make a sketch showing one method of steam-seal piping for carbon-packed glands.

45. Why should the steam-seal drains be visible from the turbine room?

46. What diametral clearance should be provided between a carbon packing ring and the shaft? What axial clearance should be allowed?

47. Explain with sketches how to refit a carbon packing ring.

DIVISION 6

STEAM-TURBINE GOVERNORS AND VALVES

124. A Steam-turbine Governor Or Speed Governor Must Be Used Whenever It Is Desired To Have A Steam Turbine Run At A Constant Speed While The Load Which It Is Driving Or Its Rate Of Doing External Work Or The Supply-steam

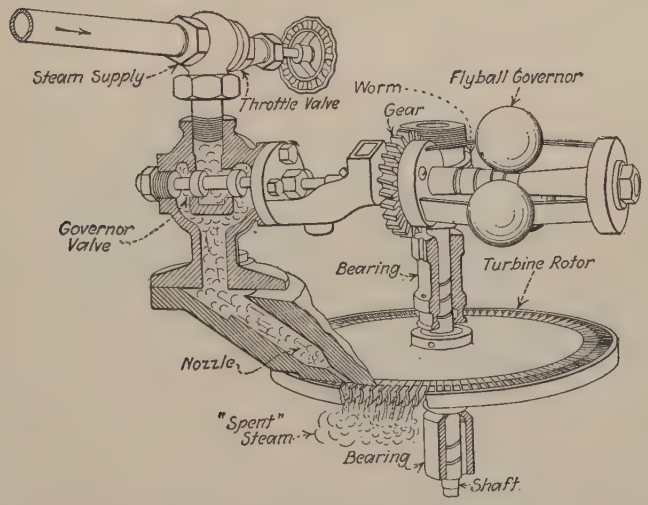


FIG. 125.—Governor used on De Laval vertical oil-purifier turbine which is of the impulse type. (*De Laval Separator Co.*)

Pressure Varies, Fig. 125 (see Sec. 27 for definition of governor). If steam were constantly admitted at the same rate to a turbine while the resistance to the turning of its rotor (due to the external load) changed considerably, its speed would fluctuate excessively. A very great load might stop it. A sudden decrease in the load would allow the speed to increase to a dangerous value. Obviously, if the speed of the turbine is to be maintained constant and unless the admission of steam is controlled by hand, there must be some automatic means of

proportioning the steam supply to the varying load on the turbine and the varying pressure of the steam supply.

NOTE.—IN MARINE SERVICE AND IN DRIVING BLOWERS, IT IS POSSIBLE TO OPERATE A TURBINE WITHOUT A SPEED GOVERNOR.—In such service, the resistance (torque) which the propeller or blower offers to the rotation of the turbine increases with the speed. The work which the turbine does increases faster than the speed. The turbine will therefore find a certain constant speed at which any given steam supply will be sufficient for the work done. In most stationary services there is a possibility that the load may be suddenly removed entirely. Then, the only limit of the turbine rotor's peripheral speed would be that equivalent to the velocity of the steam jet, which is usually high enough (Sec. 10) to burst the rotor due to the centrifugal force.

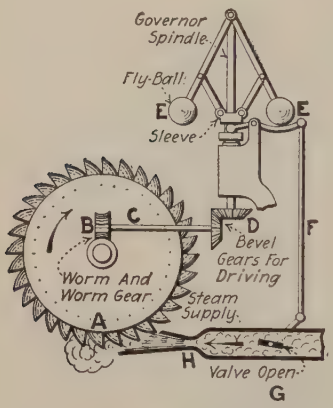


FIG. 126.—Diagram of direct throttling governor for a steam turbine. (The imaginary construction here shown is never used in practice.)

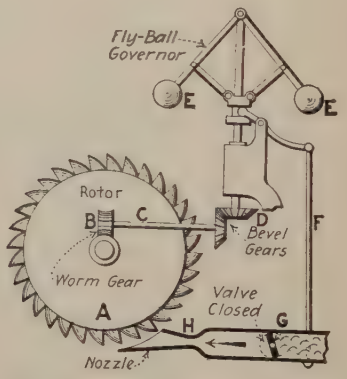


FIG. 127.—Diagram of the same imaginary governor as Fig. 126 but in the closed position.

125. How A Governor Keeps The Speed Of A Turbine Nearly Constant, in spite of considerable variations in load, may be understood by a study of Figs. 125 and 126.

EXPLANATION.—Figure 126 shows an imaginary turbine governor. The steam flows through the nozzle, *H*, and impinges on the buckets of rotor, *A*, causing it to rotate. The movement of the rotor shaft is reduced and transmitted through worm gear, *B*, shaft, *C*, and bevel gears, *D*, to the spindle of a fly-ball governor. The weights, *E*, of the governor rise due to centrifugal force (see the author's STEAM-ENGINE PRINCIPLES AND PRACTICE). The vertical movement of the weights is transmitted

through the drop-rod, F, to butterfly valve, G. If the speed of the rotor increases beyond a certain value, the weights will fly out so far that the valve will be entirely closed as in Fig. 127. Then the speed of the rotor will naturally decrease for lack of steam. The weights then fall and more steam is admitted as in Fig. 126. In this way, the governor being properly designed and adjusted, the turbine is prevented from running much faster or much slower than its rated speed.

Note.—The Speed Regulation Of A Turbine is the ratio of the speed decrease from no load to full load to the full load speed. Or, expressed as an equation:

$$(27)\quad \textit{Speed regulation} = \frac{(\textit{No-load speed}) - (\textit{Full-load speed})}{\textit{Full-load speed}} \quad \text{(decimal)}$$

126. A Complete Governing-mechanism For A Steam Turbine consists of several parts. There is always a centri-

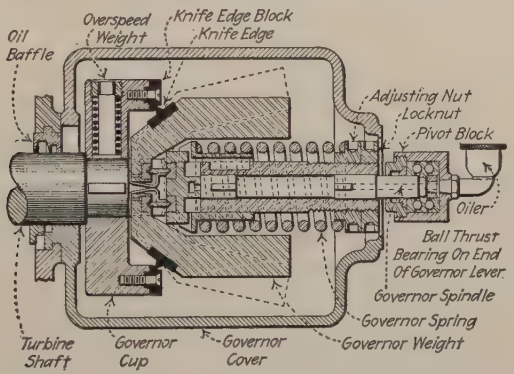


Fig. 128.—Governor of *Moore* steam turbine. (Instruction Card No. 2.)

fugal device (Fig. 128) or rotating part commonly called the governor proper. This device usually consists of movable weights so mounted that they are acted on by centrifugal force and, in some designs, by inertia also. An exception to the general construction is the pneumatic governor of the Ridgway turbine shown in Fig. 157. This governor mechanism has a pressure blower directly connected to the shaft instead of the usual movable weights. Since the pressure developed by the blower varies with its speed, the blower pressure can be used to regulate the speed of the turbine (see Sec. 148 for a description of the operation of this type of governor). There is always also a valve or a number of

valves (Fig. 129) in the steam passage leading to the turbine nozzles which valves are in some way controlled by the centrifugal force of the weights. Between these two essential elements (the governor proper and the valves) there is a connecting mechanism of some one of the many kinds which

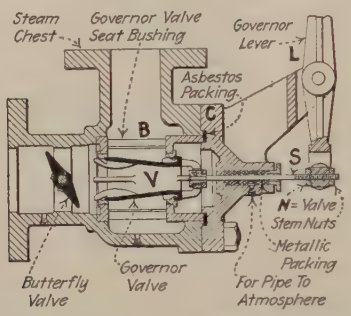


Fig. 129.—Steam chest of *Moore* steam turbine (Instruction Card No. 2). The governor valve, *V*, is operated by the governor (Fig. 128) through lever, *L*, and valve stem, *S*. Valve, *V*, is of the balanced type which has seats in a valve bushing, *B*. A steam-tight joint is made between the end of *B* and the steam-chest cover, *C*, with asbestos packing soaked in graphite and oil. Metallic valve-stem packing is used. A lantern gland is used from which the leakage along the stem can be piped to the atmosphere. The relative position of the valve is fixed when the turbine is tested and the valve stem nuts, *N*, are pinned to the shaft. This adjustment should not be changed.

are used by various manufacturers for transferring the governor-weight motion or blower pressure to the admission valves.

Note.—Steam-pressure And Oil-pressure Governor Systems are employed, as is explained later, on large turbines through pilot or relay valves. These pilot or relay valves multiply the force derived from the governor proper and are necessary because, in large turbines, the force required to move the governor valve is so great that it is impractical to operate the valve directly by the governor proper. Governors which employ such systems are called *relay governors*. Many different mechanisms are also in use which obviate part of the losses of available energy which result from throttling the steam at light loads through valves which are "cracked" or nearly closed. One of these mechanisms which admits the steam to the turbine in "puffs" is described in Secs. 136 to 138; another which admits the steam through a multi-ported valve, in Secs. 144 and 145. The term *governor* is used in the following table to indicate a complete governing-mechanism and not merely the governor proper.

127. Table Showing The Various Ways In Which Turbine Speed Governors May Be Classified and the various subclasses under each classification (see preceding note). (These classes will all be explained and illustrated in the following sections.)

Classification	Classified with respect to	Class No.	Class or description	Illustration
A	Actuating force.	1	Centrifugal.	Fig. 136
		2	Centrifugal and inertia.	Fig. 156
		3	Air pressure.	Fig. 157
B	Method of valve control.	4	Direct.	Fig. 130
		5	Mechanical indirect.	Fig. 159
			Indirect or relay:	
		6	(*a*) Steam relay.	Fig. 143
		7	(*b*) Oil relay.	Fig. 142
C	Valve arrangement or steam control.	8	Throttling.	Fig. 130
		9	Varying nozzle area (multiple valve).	Fig. 152
		10	Intermittent.	Fig. 147
		11	Bypass.	Fig. 145

128. A Direct Centrifugal Throttling Governor (classes 1, 4 and 8, Table 127) operates as explained in Figs. 126 and 127. Governors of this type are widely used on small turbines. They are very simple as compared with some of the other types and are, on the whole, very reliable. The throttling action of the control valves of governors of this type decreases the efficiency of the turbine somewhat at light loads; it is to avoid this loss in efficiency that other methods of steam-flow control are employed in governing. Some commercial governors of this type will be explained in the following sections.

129. The Main Governor Mechanism Of The Sturtevant Turbine shown in Fig. 130 represents one commercial application of a governor of the direct centrifugal throttling type. The spindle of the governor is horizontal and the movement of the centrifugal weights is opposed by a single heavy spring.

Explanation.—As the speed increases, the centrifugal weight arms (*A*, Fig. 130) "fly out," forcing the governor spindle, *B*, against the ball-bearing socket, *C*, located in the head of the bell-crank lever, *D*. This motion is in turn imparted, through the eye-bolt, *M*, to the valve, *Z*, which, in rising, closes the steam ports. When the speed decreases, the action is as follows: The weight arms, *A*, are drawn in by the centripetal force of the main governor spring, *E*. The external spring, *P*, pulls down on the end of the bell-crank, *D*, causing the ball-bearing socket, *C*, to follow the inward movement of the governor spindle, *B*, thereby lowering and opening the valve, *Z*. In other words, the governor closes the valve and the external spring, *P*, opens it. The tension of this spring,

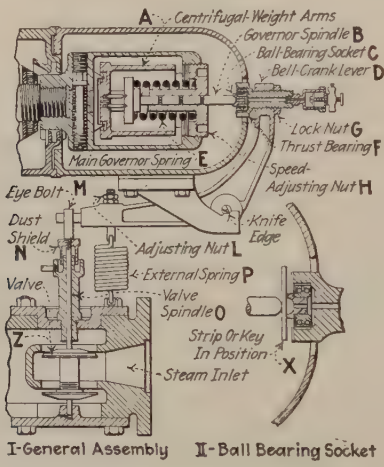


Fig. 130.—Main governor mechanism of Type-6 *Sturtevant* turbine.

P, is varied by the adjusting nut, *L*. The dust shield, *N*, prevents dust or grit from working in around the spindle, *O*, which would increase its friction.

130. To Adjust The Sturtevant Governor Valve (*Z*, Fig. 130), proceed as follows: Insert a strip or key $\frac{1}{16}$ in. thick between the governor spindle, *B*, and the ball-bearing socket, *C*, as shown at *X*. This is done with the throttle valve closed. After opening the throttle valve, the block, *X*, being in position as shown, the steam gage on the turbine steam chest should then show a slight pressure, say 10 or 15 lb. per sq. in. If there is a higher pressure than this, the valve, *Z*, should be raised by adjusting the eye-bolt *M*; in other words, remove the

bell-crank, *D*, loosen the dust shield *N* (which also acts as a lock-nut) and screw the eye-bolt, *M*, on the valve spindle, *O*. If no pressure shows on the turbine-steam-chest gage or the pressure is too low, it can be increased by lowering this valve. This setting will give the maximum opening for full load and will, at the same time, prevent overspeeding at light or no loads. The valve adjustment can also be made at thrust-bearing body, *F*, by first loosening the locknut, *G*.

Note.—The Thickness Of The Stock Used Between The Governor Spindle, *B*, And The Ball-bearing Socket, *C*, as shown at *X*, varies for different types or turbines. On turbines equipped with a ball-bearing step (*F*, Fig. 130) on the governor pin end use $\frac{1}{32}$-in. stock for types A-6 and B-6, and $\frac{1}{16}$-in. stock for types C-6, D-6 and E-6 turbines. On turbines which are *not* equipped with a ball-bearing step on governor pin end, use $\frac{3}{16}$-in. stock on all types. If the governor seems to "jump" or remains unsteady, and thus interferes with the operation of the turbine, this can be eliminated by adjusting the lower valve disc. It may be necessary to make several trials in order to determine the correct location of the valve disc. In making this adjustment the disc should not be moved more than $\frac{1}{4}$ of a turn at one time and, of course, should be securely locked after each adjustment.

Note.—Adjustment For Change In Speed Of The Sturtevant Governor (Fig. 130) may be made by adjusting the nut, *H*, in the end of the governor. The speed will be increased by screwing in the nut and lowered by backing it out.

131. A Direct Centrifugal Throttling Governor Which Is Provided With An Auxiliary Vacuum-breaker Attachment is shown in Fig. 131. When a turbine is operated condensing, it may be necessary to break the vacuum in order to prevent racing when the load is removed suddenly.

Explanation.—If the nut, *D*, which is deflected by the movement of the governor, travels outward more than about $\frac{1}{8}$ in., it engages the end, *J*, of the hollow valve stem, *T*. The movement of *T* admits air to the turbine exhaust passages through ports *O* and *P*.

132. Other Direct Throttling Governors are shown in Figs. 132, 133, 134, and 135. That in Fig. 135 is almost identical with the leaf-spring governors used for small steam engines. (See the author's Steam-engine Principles And Practice.) The following instructions for care and adjustment of these

governors may be applied to almost any small governor of the direct throttling type.

Note.—Speed Adjustments Of Direct Throttling Centrifugal Governors, to provide a speed 2 or 3 per cent. greater or less than the existing speed may always be made on governors of the coil-spring type by screwing up or slacking off on the main-spring tension. Very slight changes in speed may also be secured by varying the external spring tension. In changing the tension on either of these springs, care should

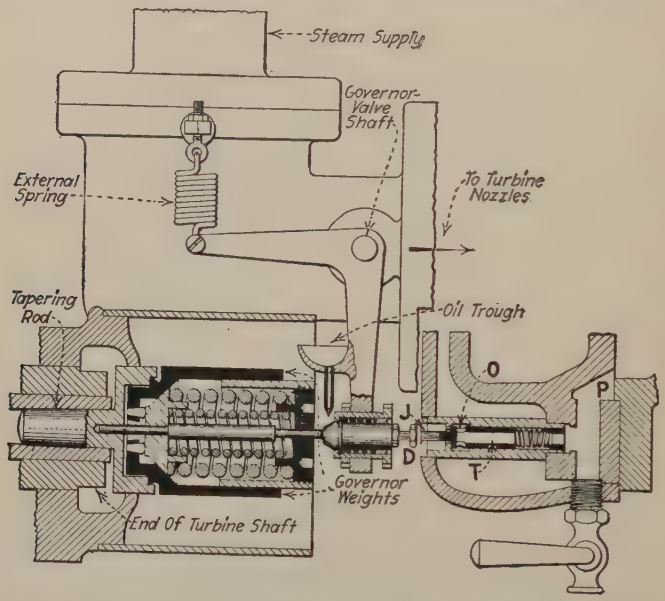


Fig. 131.—*De Laval* governor equipped with vacuum breaker.

be taken to prevent the springs from becoming "coil-bound"—entirely closed—when in service. For any material change in speed, in governors not provided with regular speed-changing handwheels, it is best to consult the manufacturers who will usually supply new weights or springs for the new speed. After any adjustment, the governor should be examined, moved by hand or watched to make sure that it shuts off at no load and moves freely in all positions.

Troubles Of Direct Throttling Governors are generally *hunting* or *racing* due to sticking of the mechanism or faulty adjustment. Lost motion will also cause hunting. Lost motion may be taken up in the valve stem (Fig. 132) of some governors. The lost motion may usually be detected by moving the various parts and observing the fit. A

certain amount of lost motion in the stationary position is sometimes recommended by the manufacturer. This lost motion must not be so great as to prevent the governor shutting off. A sticking valve stem may

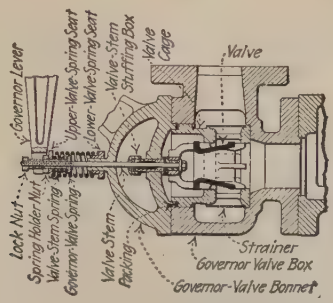


FIG. 132.—Governor valve of *Terry* turbine.

usually be detected by pushing the valve in and noting if it springs back. If the valve does not shut off at no load and thereby allows the turbine to race, it probably leaks or its stem is too short. The effective length of the stem can be increased by means of adjusting nuts. The cause of

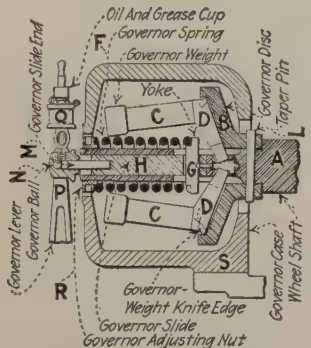


FIG. 133.—Governor of *Terry* steam turbine. (The shaft, *A*, supports the governor disc, *B*, by means of a taper shank which is keyed in position by taper pin, *L*. The governor weights, *C*, are supported on knife edges, *D*. The weights move the governor slide, *H*, outward by means of the yoke, *G*, against the tension of spring *F*. The movement of the slide is communicated to lever, *P*, by means of slide end, *M*, which revolves against ball, *N*. Oil is fed by *Q* to the ball thrust. The governor is housed in *S*. The main speed adjustment is by nut *R*.)

leaks should be investigated. If due to rust, the valve can be cleaned to insure a better seat. Conical-seated valves may be refinished on a lathe and "ground in" by an experienced machinist. Corrosion of the valve is prevented by keeping the turbine well drained when it is idle.

Note.—The Following Possible Causes Of Governor Hunting are given by the Westinghouse Electric & Mfg. Co. for the direct throttling governors on their mechanical-drive turbines. (1) Too great a travel

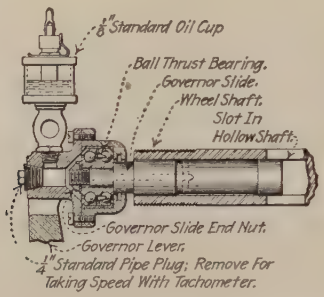


Fig. 134.—Ball thrust bearing in governing mechanism of some *Terry* turbines.

of governor poppet valve. (2) Sticking of governor poppet valve on guide. (3) Sticking of governor spindle. (4) Bent valve stem. (5) Broken governor weight knife edges. (6) Distorted or bent governor linkage. (7) Weakening of governor springs.

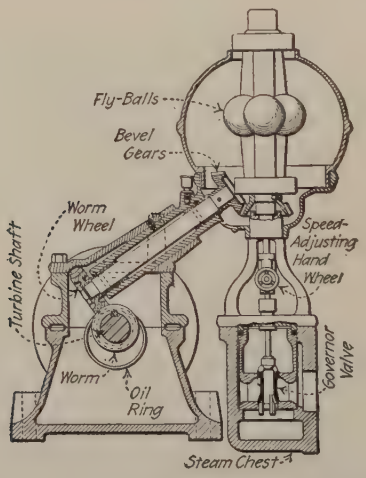


Fig. 135.—Governor of *Steam Motors Company* turbine.

133. The Emergency—Or Overspeed—Governor Mechanism Of The Sturtevant Turbine (Figs. 136 and 137) operates only in case of failure of the regular speed governor. When

the turbine is running properly, the speed is controlled or governed by the speed governor; that is, the turbine is said to be "running on the governor." But should the governor lose control of the turbine (permitting it to run too fast) there is danger of accident unless some safety device, which will act automatically, is provided to "shut down" the turbine. To

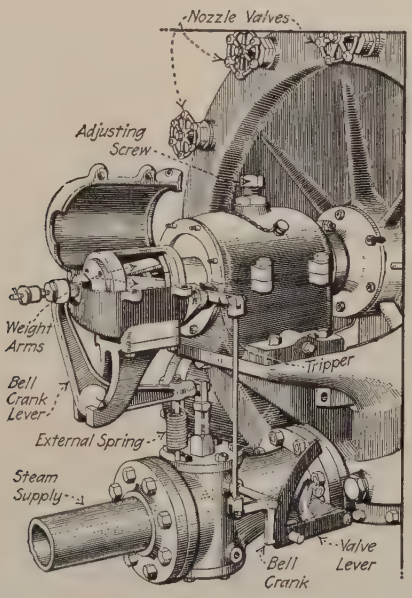


Fig. 136.—Emergency- and main-governor-mechanism assembly of Type-6 *Sturtevant* turbine.

avoid this danger, the emergency governor is provided. See explanations under Figs. 137 and 138.

134. To Adjust The Emergency Governor (Figs. 136 and 137) screw in or out on the adjusting plug, which is located, opposite the point where the piston, *A*, protrudes. Screwing this plug alters the relation of the piston's center of gravity to the center of rotation. Consequently, the closer the center of this plug is to the center of the shaft, the higher will be the speed at which the emergency governor will operate, and vice versa. Do not make the mistake of adjusting the stop bushing which holds the piston spring in position, for this will change the

distance which the piston extends when it flies out. The clearance between the tripper, *B*, and the rotating element

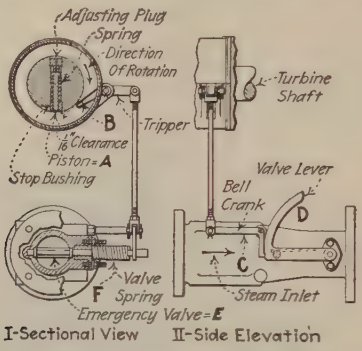


FIG. 137.—Emergency-governor mechanism of Type-6 *Sturtevant* turbine. When overspeeding, the piston, *A*, "shoots out" and strikes the tripper, *B*. *B* then causes the bell crank, *C*, to release the valve lever, *D*, which is directly connected to the emergency valve *E*, thereby causing *E*, to close. *E* is kept open by the valve lever *D* being held up by the bell crank *C* against the action of a strong valve spring *F*. When *D* is released, the strong spring comes into action, causing the rapid closing of the valve.

should not be more than $\frac{1}{16}$ in. If the emergency governor trips, it cannot be reset until the speed of the turbine has

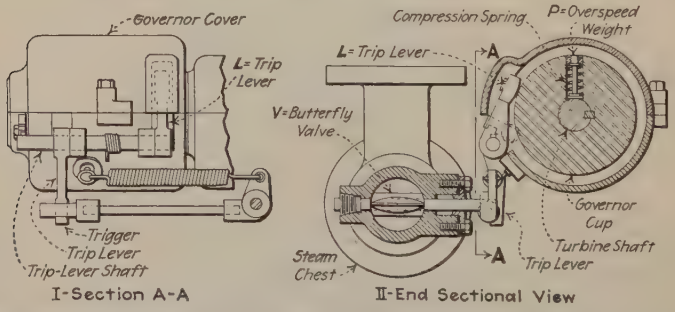


FIG. 138.—Overspeed governor, *Moore* steam turbine. (Instruction Card, No. 2.) This overspeed governor consists of a small pin, *P*, which is held in place by a compression spring. At a certain predetermined speed, for which the governor is set, this pin is thrown out and trips a latch, *L*, operating a butterfly valve, *V*, which cuts off the supply of steam to the turbine. See also Fig. 128 for another view of this emergency governor.

decreased to about one half of its running speed. This action is caused by the pin being unstable and moving to its limit when once started. The emergency governor should be

adjusted to trip at about 10 per cent. above the normal running speed. The emergency governor should be tested

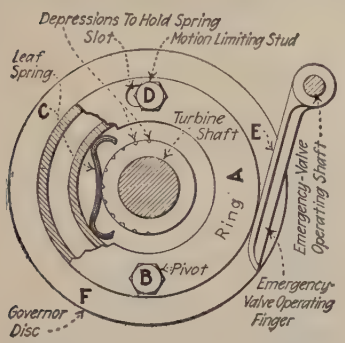


FIG. 139.—Ring-type emergency governor used on the smaller *Terry* turbines.

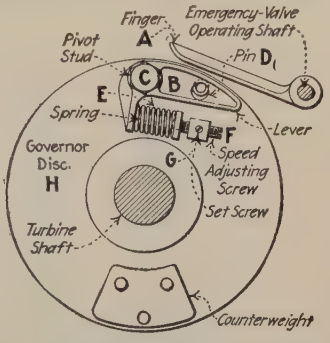


FIG. 140.—Pivoted-lever type of emergency governor on *Terry* turbines.

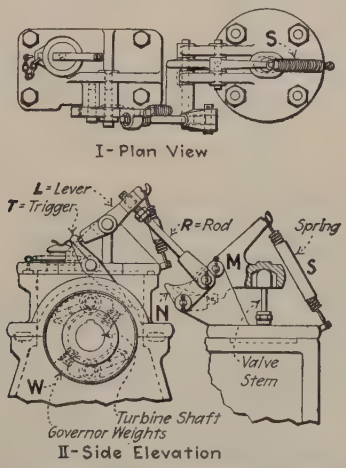


FIG. 141.—Emergency governor of *Steam Motors Company* turbine. (Steam Motors Company, Springfield Mass.) The emergency governor is a device for shutting down the machine in case of a "runaway." It is not a speed-regulating governor. The governor weights, *W*, are so adjusted that when the turbine shaft attains a speed 10 per cent above the maximum operating speed they will "fly out." They then strike trigger, *T*. This trigger releases lever *L*, which gives a hammer blow to rod *R*, releasing the other tripping mechanism on the valve bonnet. The emergency valve will then be closed by spring *S*. To reset this emergency trip, lift *M*, set *N*, in position and replace the catch *T*.

periodically by holding the governor rod against the force of the centrifugal weights, until a 10-per cent. overspeed is

obtained as shown by a voltmeter or reliable tachometer. It is important that the overspeed governor mechanism be always ready for an emergency. Nearly all emergency governors may be adjusted to trip at a lower speed by moving the weight further from the center of rotation.

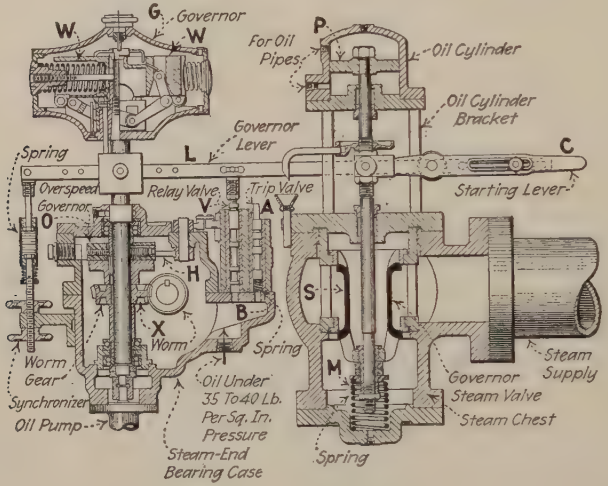


Fig. 142.—Oil-relay governor and steam chest of *Moore* steam turbine (Moore Steam Turbine Corporation, Wellsville, New York; Instruction Card No. 3). A governor, *G*, is used to actuate the oil-relay control. An increase of speed causes the weights, *W*, to move outward. This moves lever *L* upward, moving oil-relay valve, *V*, which admits oil below piston, *P*. This causes the governor steam-valve, *S*, to close. Movement of *S* moves compensating lever, *C*, which brings *V* back to its neutral position. This stops the flow of oil and prevents over travel of the steam valve. The governor steam valve, *S*, is provided with a spring, *M*, at the lower end of its valve stem. The purpose of this spring is to automatically close the valve and shut down the turbine in case the oil pressure fails. The overspeed governor, *O*, is carried on the governor shaft above the worm wheel, *X*, which drives the governor; a weight is held in place by a compression spring until a predetermined speed, for which the overspeed governor has been set, is reached. Then the overspeed governor is thrown outward and strikes a lever, *H*, which trips a latch, allowing auxiliary valve, *A*, to be forced upward by spring *B*. This admits full oil pressure under piston *P* and exhausts oil from above the piston, closing the governor steam valve, *S*.

Note.—In Maintaining The Emergency Governor (Fig. 137) the following should be observed. The piston, *A*, should "shoot" out at a speed about 10 per cent. greater than the rated speed of the turbine. This piston should occasionally be tested for free movement. To make this test, push a wire through the hole in the center of the adjusting plug; it should be possible to thus push the piston out approximately ¼ in. It is very important to *have this piston working freely*, and a little oil applied occasionally—say once a month— will assure this free movement.

NOTE.—OTHER MAKES OF EMERGENCY GOVERNORS are shown in Figs. 138, 139, 140 and 141. Their actions and functions are similar to those already described. In general, the emergency governor should be entirely independent of the speed governor.

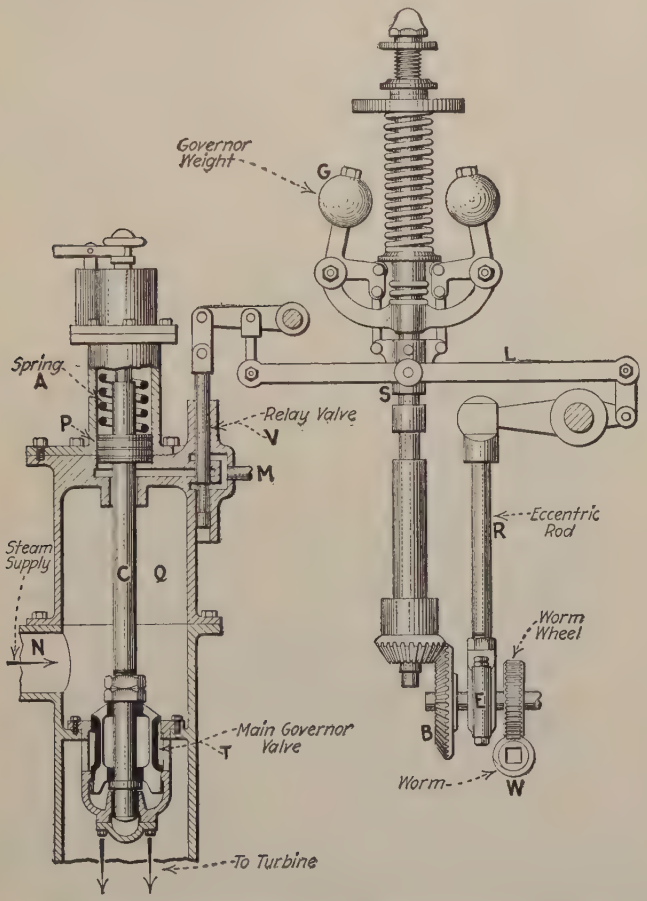


FIG. 143.—Diagram showing operation of the older-type Parsons turbine governor.

135. An Oil-relay Throttling Governor (Fig. 142), accomplishes the same result as does the direct throttling governor but does not depend on the centrifugal force of the weights to operate the main governor valve. Instead, the centrifugal force of the weights operates a small valve which admits oil above or below a piston the rod of which controls the main valve.

136. Centrifugal Steam-relay Intermittent Or "Blast" Governors are used on a large number of Westinghouse and foreign Parsons turbines. Oil-relay governors (Sec. 138) are superseding this type. The principle of operation of the governor may be understood by examination of Fig. 143. Its action is, briefly, to admit steam to the turbine nozzles in "puffs," the length of the "puff" depending on the load. The "puffs" occur at regular intervals and so frequently

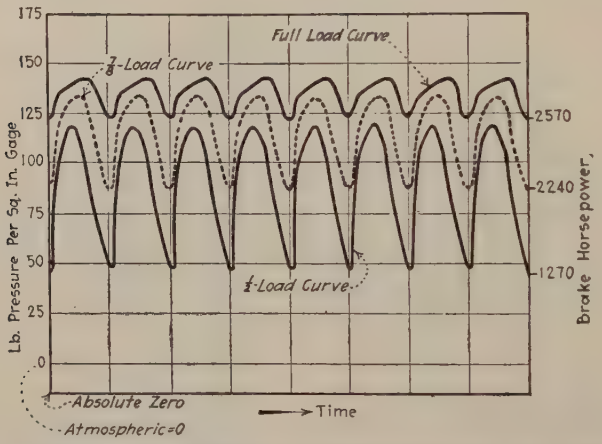


Fig. 144.—Graphs showing the effect of an intermittent governor on the instantaneous steam pressure in turbine live-steam parts.

that there is no uneven effect on the speed of the turbine. The principal object of this action is to have the valve either entirely closed or wide open most of the time, so that there will be little throttling. Another advantage is that, since the valve is constantly moving, the possibility of its "sticking" is minimized. With the advent of the larger turbines this "puff" system of admitting steam was found to cause, at times, objectionable vibration in the main steam lines of the power house. About 1909 the steam relay began to be abandoned for the oil-pressure-relay system.

Explanation.—The turbine shaft (Fig. 143) carries a worm, *W*. The shaft of the worm wheel which engages *W* carries an eccentric, *E*, and a bevel gear, *B*, which drives the spindle of the centrifugal governor, *G*. There is a system of levers connected to the eccentric rod, *R*, through

which it gives a reciprocating motion to the plunger of the relay valve, V. The live steam is admitted at N, flows through the space, Q, around the piston rod, C, and lifts the piston, P, which controls the governor valve, T. This allows steam to flow through T to the turbine as long as the valve, V, is closed. But when V is open, the steam escapes at M (into the engine room) faster than it enters at Q; thereby the piston is forced down by the spring, A, which presses behind it. One of the levers, L, is pivoted on the sleeve, S, of the governor so that when the governor lifts, V moves between higher limits and allows steam to escape at M for a longer period. In this way, the valve, T, is made to remain closed longer when the speed of the turbine is higher. The effect of this action on the steam pressure is shown in Fig. 144.

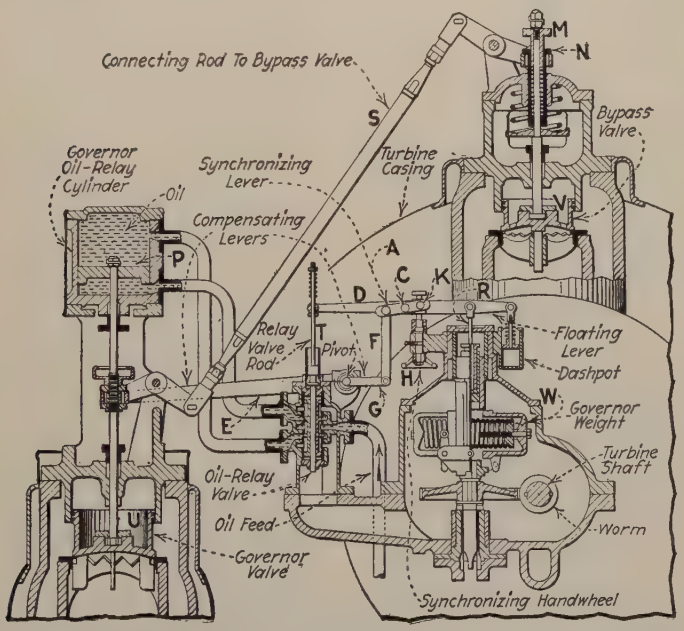


Fig. 145.—Throttling and bypass governor used on *Allis-Chalmers* reaction turbines. See Fig. 146 for an enlarged view of the oil-relay valve.

137. An Allis-Chalmers Oil-relay Throttling And Bypass Governor which is used by that company on 5,000 to 15,000 kw. turbo-generators is shown diagrammatically in Fig. 145. Its action is similar to that already described for oil-relay governors in Sec. 135 except for the bypass and synchronizing devices.

Explanation.—As the turbine speed increases, weights W fly outward and raise the vertical rod, R, which is attached to the floating lever, D. This lever D, being supported at pivot C, pushes down on the stem, T, of the oil-relay valve (Fig. 146) thus opening its ports so that oil pressure is admitted abovepiston P. This closes the governor valve, U. But as U moves down, it moves compensating levers, E and G, and thereby moves upward F and synchronizing lever, A, which is pivoted at the fixed point K. Lever A is attached to D by pivot C. This movement in turn raises T and closes the relay-valve ports.

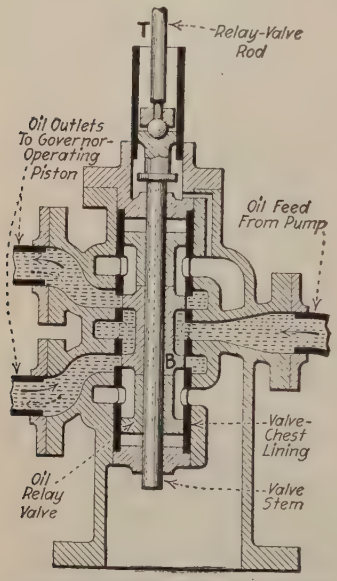


Fig. 146.—Enlarged view of the *Allis-Chalmers* oil-relay valve shown in Fig. 145.

When the turbine speed falls, due to an increased load, the above processes are reversed and the valve U is lifted from its seat. Its motion is communicated through connecting-rod, S, to the sliding collar, N. At a certain position of U, the sliding collar strikes the fixed collar, M, and the bypass valve, V, is lifted. This admits live steam to an intermediate stage of the turbine. Thus the bypass valve remains entirely closed at light loads and opens for heavy loads. The end K of the short synchronizing lever, A, which is pivoted at C to the floating lever, D, may be raised by screwing up on the handwheel, H. This changes the position of the relay valve with respect to the main governor valve and so changes the speed of the turbine. A 5-per cent. regulation above or below normal speed may thus be obtained.

Note.—Bypassing Is Employed In Many Large Modern Multi-stage Turbines as a means of carrying overloads. The steam which is bypassed to a later stage of the turbine is not used with as high an efficiency as that which flows through all of the blading. There is therefore, at overloads, a loss in efficiency due to bypassing but this loss is offset by the increased ability of the turbine to carry peak loads. Thus, for example, a turbine which operates at its best economy at 5,000 kw. can readily, by bypassing, be made to carry 7,000 kw. But when carrying 7,000 kw., its economy is not as good as when it is carrying 5,000 kw.

138. The Westinghouse Type Of Centrifugal Oil-relay Intermittent Governor is shown in Fig. 147 and the valves

which it actuates in Fig. 148. (Based on *Westinghouse Electric & Mfg. Co's.* Instruction Book No. 5,171.) In general, the functions of this governor (the details of operation are given below) are: (1) *To provide a throttle valve, O* (Fig. 148), *which will be controlled by the governor proper for maintaining a constant turbine speed from no load up to about full load.* This is effected by means of an oil-relay system, similar to that already explained in Fig. 142. (2) *To provide an overload bypass valve, P* (Fig. 148), *which opens at about full load and admits additional steam to a later stage of the turbine to carry overloads* as explained in the preceding section. (3) *To provide a continuous reciprocating motion of the throttle valve, O, and the bypass valve, P, when the latter is open and of the operating linkage, whereby: (a) Sticking due to starting friction is avoided. (b) Energy loss due to throttling of the steam at very light loads is avoided. Z* (Fig. 147) is the governor proper whereby the steam flow to the turbine blading is controlled by governor valves, *O* and *P* (Fig. 148), which are, as will be explained, actuated by oil under pressure as regulated by the relay-valve system, *FE*.

Explanation.—The worm, *W* (Fig. 147), mounted on the turbine shaft, drives a worm wheel which is mounted on the governor spindle. The governor proper is thus rotated. The cam, *X*, is driven by a gear on the governor spindle. This cam gives a rocking motion to the short lever, *N*, which is pivoted at *q* on the governor lever. In this way a short regular reciprocating motion, for reasons previously indicated, is transmitted through the linkage, *MYSJ*, to the oil-relay valve, *E*. See Fig. 149 for an enlarged view of this valve. As the governor raises it rotates lever *f* around its pivot *e* and hence lowers the rocking-lever pivot *q*. This causes the cam, *X*, to move the relay valve, *E*, between lower positions.

This oil-relay valve acts similarly to a piston slide valve for a steam engine. When raised it admits oil, from the pressure chamber, *H* (Fig. 149), to the under side of the operating piston, *F*, simultaneously allowing oil to flow from the upper side of the piston to the exhaust passage, *I*. When *E* is lowered, its action is the reverse and the oil is admitted above and exhausted below the operating piston. The floating lever, *G*, to which the stems or rods of both the oil-relay valve and the operating piston are attached, operates to stop the oil flow as soon as the operating piston has moved a short distance. This lever is arranged in this way so that the operating piston will not move its entire stroke for only a small movement of the oil-relay valve. It is desired that the movement

of the piston be proportional to (but much greater than) the movement of the relay valve.

The operating piston, *F* (Fig. 148), controls (by the movement which it derives from the oil pressure through the pilot valve as explained above) the two valves—a primary or governor valve, *O*, and a secondary or bypass valve, *P*. The levers, *m* and *n*, which connect the operating piston to the valve are similar except that the lever, *n*, is provided at *R* with an adjustable amount of lost motion so that valve, *P*, will not lift until valve, *O*, is open (see Sec. 141 for adjustment). The valves are

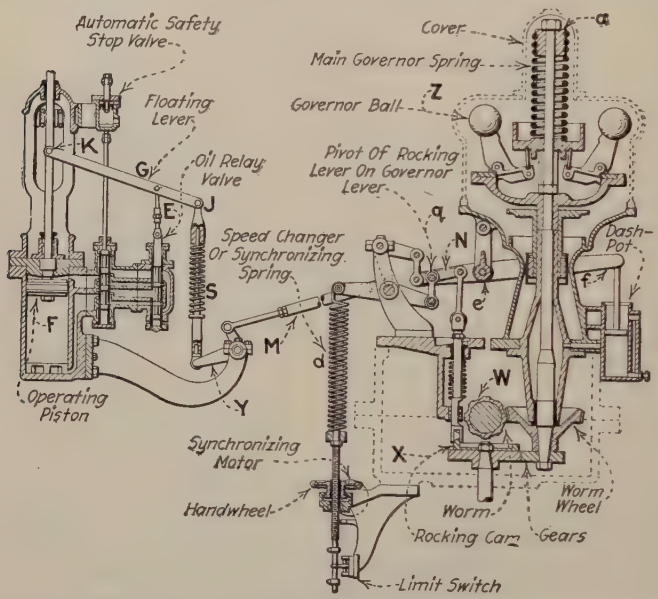


Fig. 147.—Operating gear and governor proper of a *Westinghouse* intermittent governor. The valves which this governor controls are shown in Fig. 148.

provided with main springs, *C*, which close them if the oil pressure fails — as for instance when the turbine is stopped. When shutting these valves, the governor tends to raise the operating piston and would, when the governor is not revolving, strain the linkage if it were not for the weak spring, *S* (Fig. 147). This spring is inserted in the connecting link so as to permit closing the governor valve without straining the linkage.

Note.—An Automatic Stop Valve, *Q* (Fig. 149) is provided to shut the governor valve in case of failure of the governor linkage. This valve consists of a piston, *L*, held to the top of a small cylinder by the steam pressure on its unequal upper and lower faces. Live steam is admitted at *U* above the piston but leaks past and establishes a pressure in the lower part of the cylinder as long as the opening, *V*, is closed. The

opening, V, is connected to the emergency governor (Fig. 150). When the emergency governor is tripped, it releases, through a pipe, the pressure in V and the live steam at U then forces the piston, L, to the bottom of its cylinder against its spring. The movement of L throws a piston valve, T, which operates just as does valve, E, to close the governor valves.

139. To Check The Adjustment Of The Westinghouse Centrifugal Governor (Z, Fig. 147) first adjust the speed changer spring, d, so that it will have practically no tension when the governor balls or weights are in their innermost position. The main governor spring (which is held by nut, a) should now be adjusted so that the turbine will run at 5 per cent. below normal speed at no load. Then tighten d until the speed of the turbine is normal. There should now be the proper amount of speed regulation—about 1 per cent. between no load and full load. If there is not, then, for less speed regulation, adjust the nut, a, so as to render more coils of the main spring effective; for more speed regulation, so adjust that fewer of the spring coils are effective.

Note.—Speed Adjustments While The Turbine Is Running are made by means of the spring, d. The wheel which tightens or loosens this spring may be so arranged as to be turned by a motor, which is controlled from the switchboard, so that the turbine may be synchronized with another one for parallel operation.

140. The Oil-relay Control Adjustment Of The Westinghouse Oil-relay Intermittent Governor (Figs. 147 and 149) should be made after the governor proper has been adjusted, as described in the preceding section. The method is as follows: With the oil-relay control connected and the oil pressure established, permit the turbine to turn slowly under steam so as to make lever, N, oscillate. The governor balls or weights should be in their innermost positions. Manipulate the oil-relay valve, E, by holding down on the pivot, J, to bring operating piston, F, into mid-position. Then adjust link, r (Fig. 149), so that when oil-relay valve piston, E, is in mid-position and will not admit oil either above or below the operating piston, F, the lever, G, will be horizontal. Then release J so that the spring link, S, is at its full operating length (not compressed) and the piston F, will move to its

extreme bottom position. Now adjust link, *M*, until the piston, *F*, has a slight movement. Finally, lengthen *M* by giving it one and one-half turns.

141. The Setting Of The Primary And Secondary Intermittent-governor Valves Of The Westinghouse Turbine (Fig. 148) may be checked as follows: The amount of travel of the valves from their extreme positions to their mid-positions, when the levers *m*, and *n*, are horizontal, should be noted

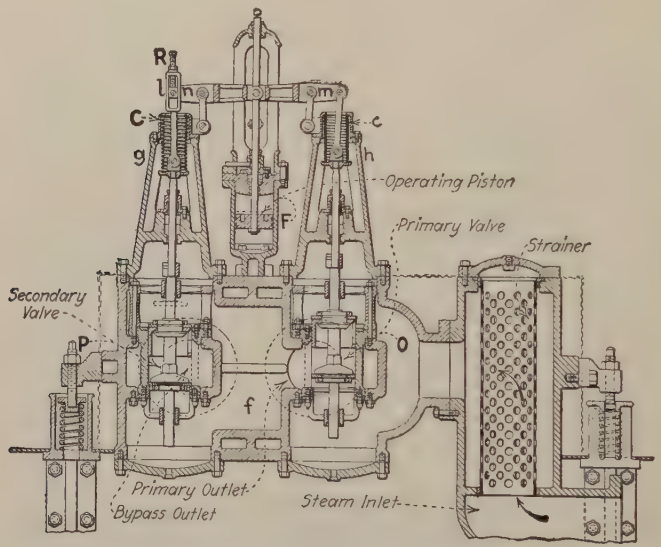


Fig. 148.—*Westinghouse* operating cylinder, primary and secondary valves controlled by the governor of Fig. 147.

at the time the turbine is delivered as complete by the erector. These travels should be afterwards maintained. With the primary valve, *O*, just leaving its seat, the piston, *F*, should be ½ in. from the end of its stroke. This may be adjusted by inserting liners at point, *h*. When the piston, *F*, is in its extreme upper position there should be from 1/32 to 1/16 in. clearance underneath link block, *l*. This may be adjusted by inserting liners at point, *g*. The adjusting screw, *R*, should be so adjusted that the secondary valve, *P*, will open at the moment the primary valve, *O*, reaches its maximum port opening, as shown by the pressure in the space, *f*.

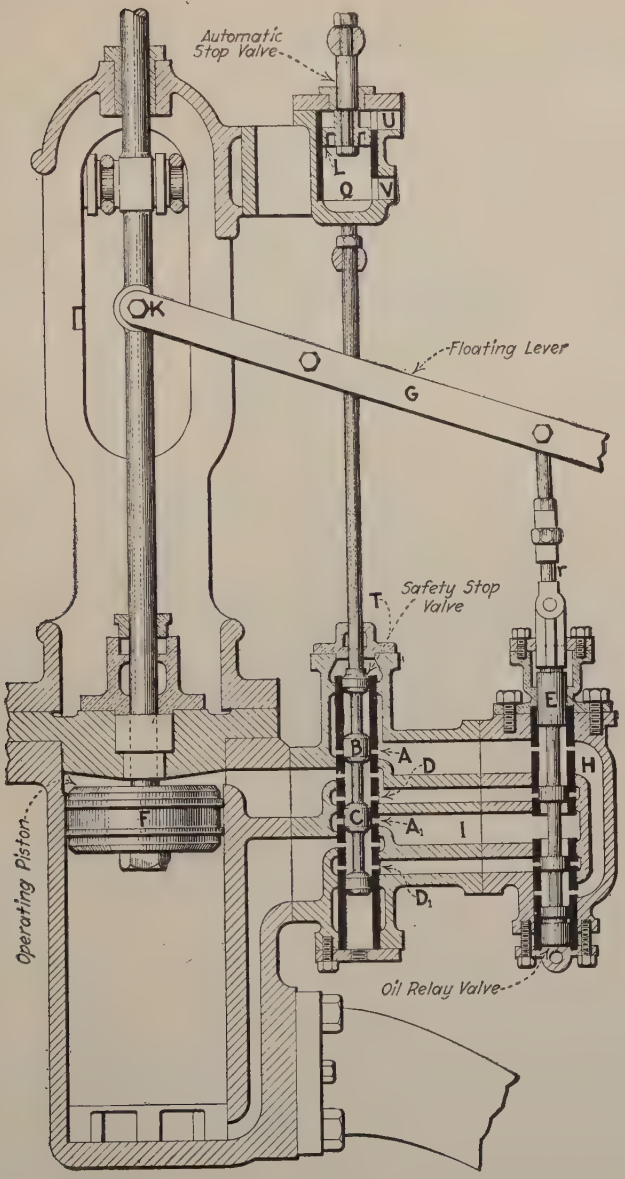


Fig. 149.—Enlarged view of the *Westinghouse* relay valve of Fig. 147.

142. The Automatic Stop Adjustment Of The Westinghouse Intermittent-governor Turbine may be checked as follows: With the automatic stop piston, L (Fig. 149), at the upper end of its stroke, the enlarged parts, B and C, of the safety stop plunger, T, should be central over the ports, A and A_1. With the automatic stop piston, L, at its lowest position, the enlarged parts of the safety stop plunger, T, should be central over the ports, D and D_1.

143. A Westinghouse Safety Stop Or Emergency Governor (Fig. 150) releases steam pressure in a pipe when it trips and

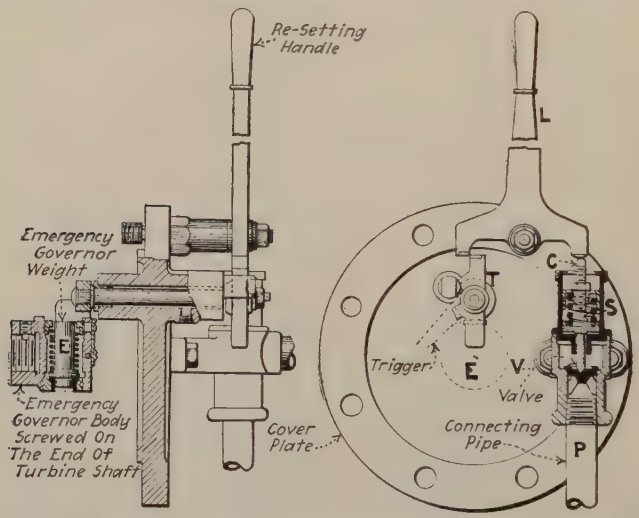


Fig. 150.—Automatic emergency governor or safety stop which is used on some Westinghouse turbines in connection with the throttle valve of Fig. 151 and the valve, T, of Fig. 147.

this drop in pressure operates one or more automatic valves in other parts of the turbine.

Explanation.—The weight, E, flies out at the speed at which the emergency governor is set to operate and trips the trigger, T. This allows the spring, S, to force the lever, L, free of the set screw, C. The steam in the pipe, P, then raises the valve, V, and escapes so that the pressure in P falls. The steam is thus allowed to escape from opening J (Fig. 151) of the automatic throttle valve and from the opening (V, Fig. 149) of the safety governor valve, so that both the throttle and the governor valves are closed (see Sec. 138 and caption to Fig. 151) whereby the steam supply to the turbine is cut off.

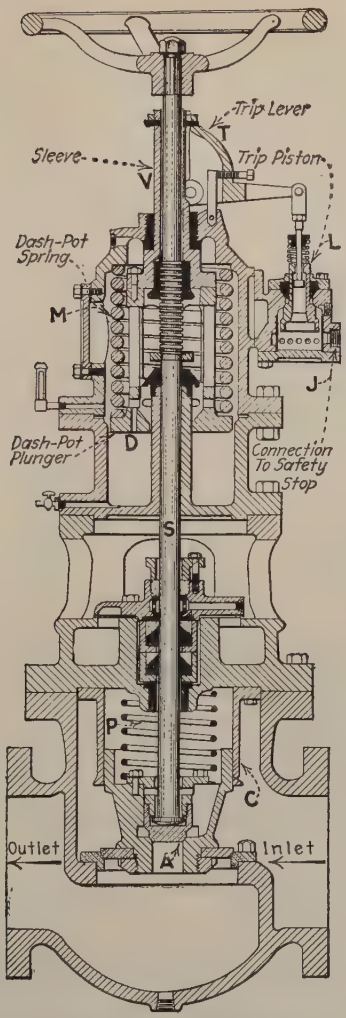


Fig. 151.—*Westinghouse* automatic throttle valve which is used in connection with the safety stop or emergency governor of Fig. 150. (So long as the emergency governor does not release the pressure at *J*, the valve may be operated as a common throttle valve. The pilot valve, *A*, and cylinder, *C*, balance the valve to assist in opening. The spring, *P*, prevents chattering. When pressure is released at *J*, the trip piston, *L*, moves due to the live-steam pressure behind it, and trips the lever, *T*, allowing the sleeve, *V*, to fall. The dash-pot spring, *M*, then closes the valve. Too rapid movement of the valve is prevented by the oil dash-pot and plunger *D*. The valve may be re-set by turning the hand wheel at its top until the sleeve, *V*, is lifted sufficiently that the trip lever, *T*, may be put in place.)

144. A General Electric Co. Multi-ported Valve Governor is shown in Fig. 152. The steam for the turbine is admitted to the space, *S*, through the strainer, *T*. There are shoulders (not shown) on the valve stem which are so arranged that,

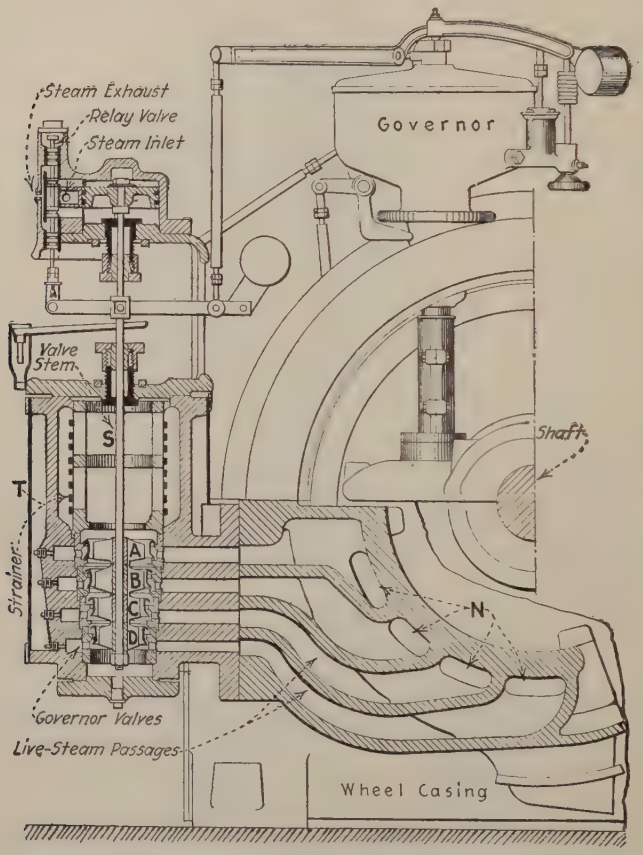


Fig. 152.—Section of multi-ported governor valve used on some *General Electric Co.* Curtis turbines.

as the valve stem lifts, the valves, *A*, *B*, *C*, and *D* are opened successively so that only one valve is opening at a time. The rest are all either closed or open. The various valves admit steam to the various nozzle passages, *N*. Thus there is very little throttling action and the governing is accomplished

chiefly by varying the number of nozzles to which steam is admitted.

145. A General Electric Co. Multiple-valve Governor Mechanism is shown in Fig. 153; this figure shows in section one of a number of similar valves which are arranged side by side along the top of the turbine casing. The governor proper (shown in Fig. 154) operates an oil-relay valve (*V*, Fig. 155) which admits oil against an operating piston. This piston moves a rack, *R*, which engages a pinion, *L*, on the shaft (*S*, Figs. 153 and 155). On this shaft are a number of cams, *C*, keyed at different angles. Thus when the operating piston moves, the cams strike successively their cam-following rollers, *R*, and lift the various poppet valves, *V*, in turn. These valves admit steam to the various nozzles and bypasses of the turbine.

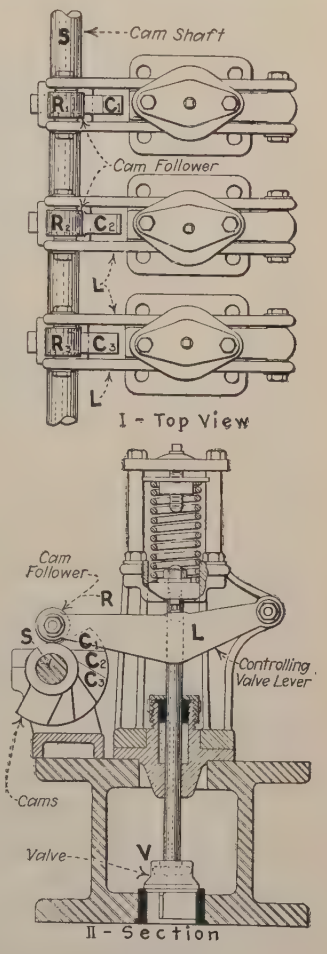


FIG. 153.—Controlling valve used for some *General Electric Co.* Curtis turbines. These valves are controlled by the governor proper shown in Fig. 154 through an oil-relay valve and rack-and-pinion device.

146. Speed Adjustments Of One Or Two Per Cent. In Spring-opposed Governors such as that shown in Fig. 154 (*General Electric Co.* INSTRUCTION BOOK No. 82,207) may be made by varying the tension on an external spring. This governor is used with the relay valve of Fig. 155 and the valve gear of Fig. 153. Governors of this sort are provided with auxiliary springs, *A*, for varying the speed in synchronizing. If it is desired for any reason to permanently change the speed at which the governor operates,

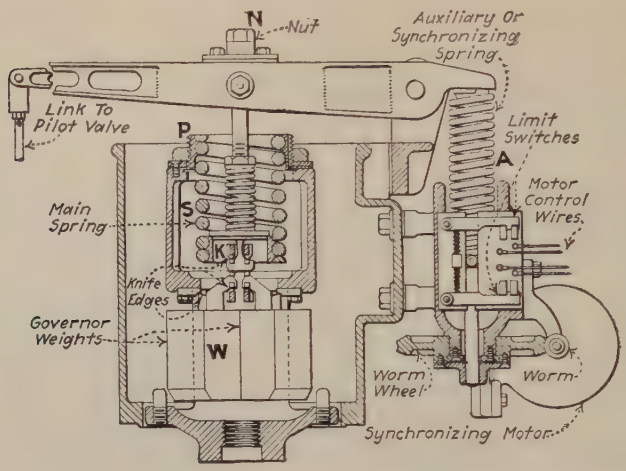


FIG. 154.—Vertical centrifugal governor used on large-capacity *General Electric Co.* Curtis turbines.

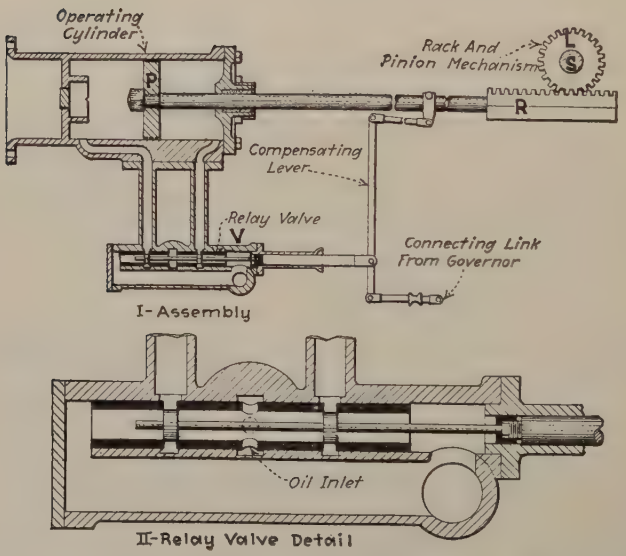


FIG. 155.—Rack-and-pinion mechanism and hydraulic cylinder used for operating governor cams on large *General Electric Co.* Curtis turbines.

this should be done by adjusting the nut, N, on the top of the governor. Adjusting N will, without affecting the speed regulation, change the speed only through a comparatively small range, on either side of that speed at which the governor was designed to operate. Too much adjustment of N will affect the speed regulation. If it is necessary at any time to increase or decrease the speed regulation of the governor, this can, within very narrow limits, be accomplished by inserting more lead—adding weight—in pockets (not shown) in the weights, W, to diminish the regulation. To increase the regulation, take lead out. However, if a considerable increase or decrease in regulation is required, it should be secured by respectively decreasing or increasing the number of working coils in the main spring, S, by screwing the top spring plug, P, in or out. A quarter turn of the plug will effect a material change in the speed regulation.

Note.—The Positive Action Of Any Governor Is Necessarily Dependent Upon The Absence Of Friction From Its Moving Parts. All knife edges, K, (Fig. 154) and joints should, if wear causes any appreciable deterioration, be renewed. In order that wear may be minimized, the governor should be assembled in such a manner that all of its rotating parts run as nearly concentric as is possible.

147. A General Electric Co. Governor Proper Which Employs Inertia And Centrifugal Force As Governing Forces is shown in Fig. 156. The two inertia arms, A, carry the centrifugal weights, W, and the inertia weights, I. As the speed increases the centrifugal weights fly out against the tension of the spring. The arms are affected by inertia and prevent sudden change in speed. The horizontal movement of the arms is changed to a vertical movement by two toggle levers, T, which fit into ball sockets on the arms.

Note.—To Increase The Turbine Speed With This Governor (Fig. 156) without changing the speed regulation, subtract weight from the weight socket, W, or *vice versa*. The weight of opposite weights, W, must be kept equal to prevent unbalancing the governor. Increasing the main spring tension increases the speed and also decreases the speed regulation, and *vice versa*. Shortening the effective spring length by screwing the plugs, P, closer together increases the speed regulation, and

vice versa. The governor is adjusted at the factory and need not, ordinarily, be altered except by the external adjustment (not shown) which is provided for the purpose.

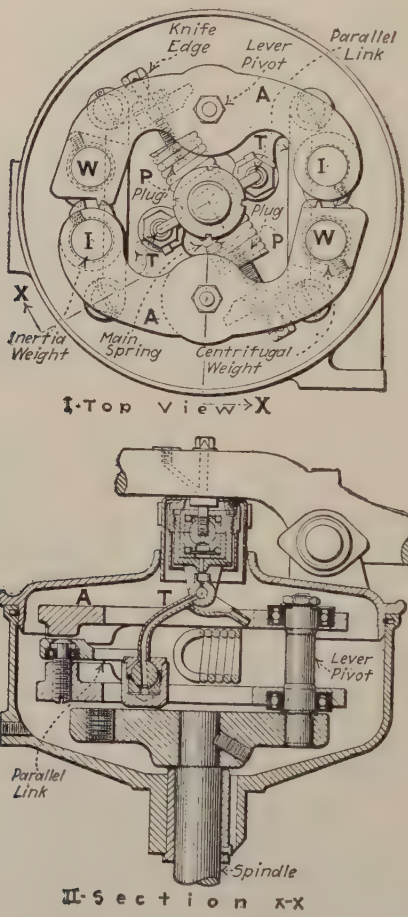


FIG. 156.—Inertia governor used on medium-capacity Curtis turbines. (*General Electric Co. Bulletin.*)

148. An Air-pressure Or Pneumatic Governor Used On The Ridgway Steam Turbine (Fig. 157) employs an air-pressure blower, *B*, (directly connected to the shaft to furnish the operating power for the governor) instead of employing

the centrifugal force developed by weights as do most governors. The blower creates an air pressure which is approximately proportional to the square of the speed. This pressure is

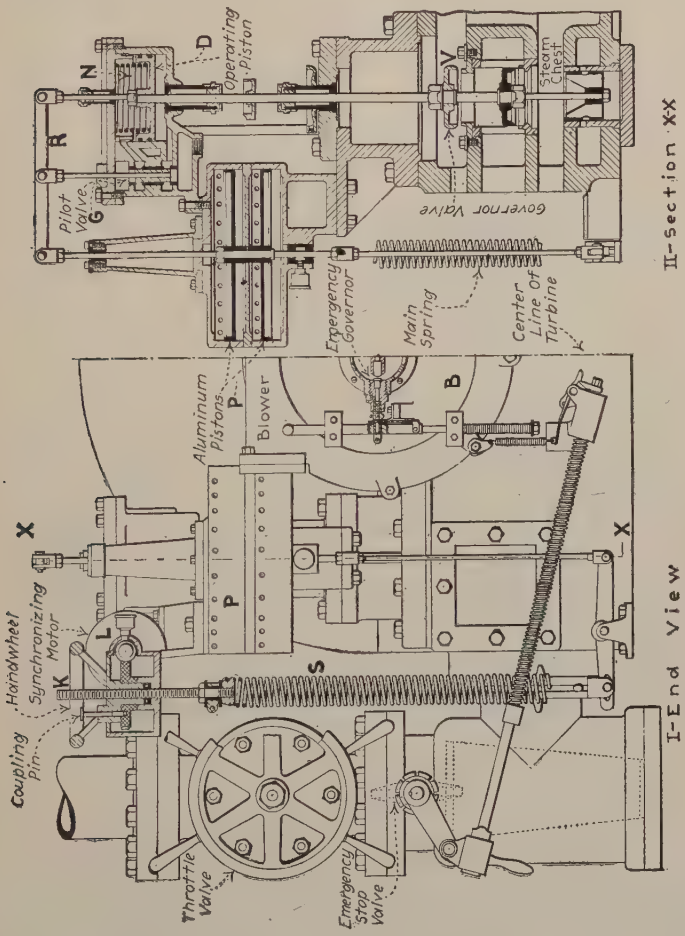


Fig. 157.—Pneumatic governor of the *Ridgway* turbine.

exerted on the under sides of two light aluminum pistons, *P*, the movement of which is opposed by a spring, *S*. The tension on this spring is varied by the handwheel, *K*, or by the synchronizing motor, *L*. The double beat throttle valve, *V*, is controlled by the operating piston, *D*, through the oil-relay

valve, G, and the floating lever, R, in the usual manner. There is a spring, N, which closes the valve in case of failure of the oil pressure. The chief advantages claimed for this method of governing are simplicity and absence of any high-speed parts on which there is friction. The runner of the blower has no friction except that of the air.

149. A So-called Mechanical Indirect Centrifugal And Inertia Governor Valve Gear (Figs. 158 and 159) is in use on some medium-capacity (say 500 kw.) General Electric Co. Curtis turbines. The illustrations show only one valve mechanism; on a turbine there are a number of duplicate mechanisms mounted side by side, all controlled by a single gover, nor and each admitting steam to or cutting it off from one nozzle section. This valve gear operates (see explanation below) in a way somewhat analogous to a detaching Corliss-valve mechanism for steam engines. That is, it employs two pawls or "pick-up hooks," A, for each valve. The pawls are attached to K and are oscillated up and down by the motion transmitted to K by L. The upper hook, A_o, opens and the lower hook, A_c, closes the valve. The position of the shield plate or "knock-off cam," E, is controlled by the governor and determines the height to which the valves, V (Fig. 159), are lifted. Unlike the Corliss mechanism, however, each valve is closed by a pawl, A_c, instead of being closed by springs or vacuum.

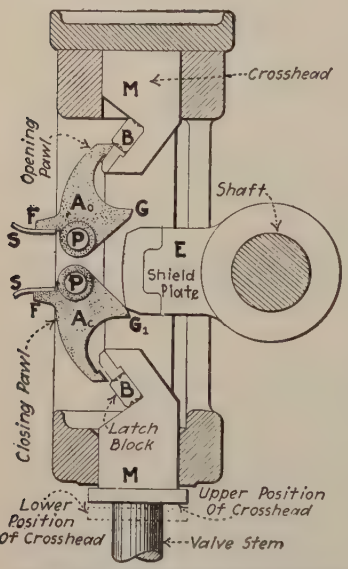


FIG. 158.—Lifting and knock-off mechanism of the *Rice* mechanisal valve gear. (*General Electric Co.*)

EXPLANATION.—The lever, K (Figs. 158 and 159), is oscillated up and down by an excentric and the rod, L, at the rate of 120 complete strokes per minute. The pawls, A, are pivoted at P and P on the

lever, *K*. Due to the tension of springs, *S*, on lugs, *F*, the pawls tend to engage the latch blocks, *B*, so as to carry the governor valves, *V*, up and down also. But the position of the shield plate, *E*, is controlled by the governor. It allows the governor valves to be lifted when the turbine requires more steam. Also when the turbine requires more steam, it prevents the valves from being closed on the return stroke. When less steam is required, the shield plate is so moved by

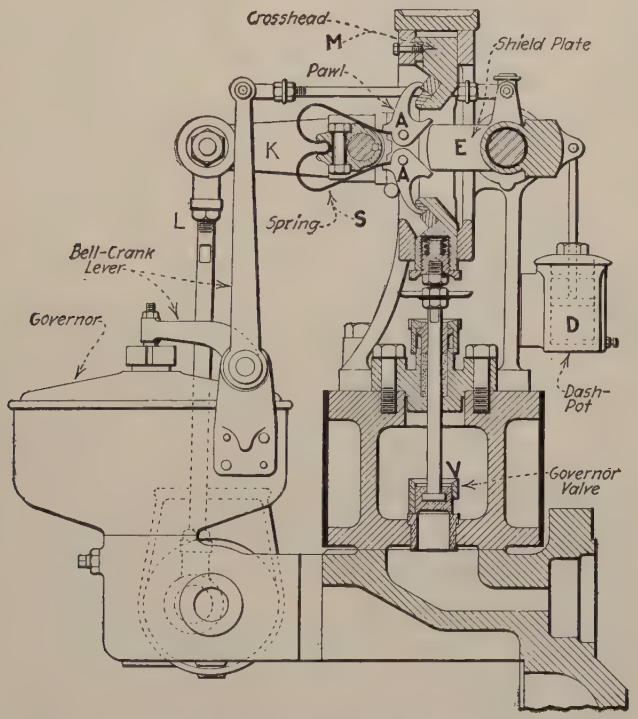


FIG. 159.—Rice mechanical valve gear used on some medium-sized *General Electric Co.* Curtis turbines.

the governor as to allow the governor valves to be closed and it similarly prevents them from being opened.

150. Dash-pots (*D*, Fig. 159) are used on many turbine governors to prevent *hunting*. If a large centrifugal governor were so adjusted as to allow a regulation of only 1 to 1½ per cent. in the speed of the turbine, the governor would have a tendency to vibrate slowly—or to move above and then below

its correct position. A dash-pot is therefore frequently used to "dampen" such vibrations of the governor and to maintain it in its correct position.

NOTE.—TO MAKE A GOVERNOR MORE SLUGGISH, or slow-moving, use a heavier oil in the dash-pot, or restrict the opening around the plunger. To make it more prompt, give it more opening or thin the oil with kerosene.

NOTE.—For more complete directions for the care, construction and adjustment of dash-pots, see the author's STEAM-ENGINE PRINCIPLES AND PRACTICE.

151. In Adjusting A Governor To Synchronize Steam Turbo-alternators, a motor-operated device which is con-

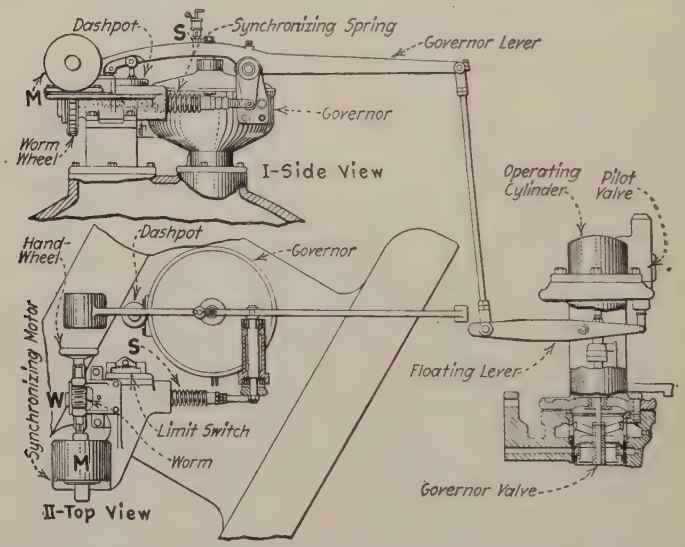


FIG. 160.—A *General Electric Co.* synchronizing device for turbo-generators which may be controlled from the switchboard. When the motor, *M*, is connected into circuit at the switchboard, it turns worm, *W*, and tightens or loosens synchronizing spring, *S*, depending on the direction in which the motor is caused to rotate.

trolled by the switch-board operator is often employed. If a turbo-alternator is to be connected in parallel with another which is already running under load, it is necessary that, at the instant of connecting the one in: (1) *The two machines be running at exactly synchronous speed.* (2) *The two machines be delivering the same voltage, as shown by a*

voltmeter. (3) *The two machines be in phase.* The "dead" machine, which is to be connected in, is usually synchronized with the "live" machine, which is already under load, by altering the speed of the dead machine until its speed is exactly the same as that of the live machine and the two are in phase.

Note.—To Adjust The Speed For Synchronizing: On the smaller turbines, this may be effected by hand adjustment of the speed—changing the synchronizing spring (*d*, Fig. 147 and *S*, Fig. 157). On the larger turbines, the speed alteration is accomplished by a motor-controlled synchronizing device (Figs. 157 and 160) which forms part of the governor. The synchronizing motor may, in order to change the speed, vary the tension of the governor synchronizing spring as in Figs. 157 and 160 or it may change the position of the pilot valve with respect to the governor valve as in Fig. 145. In Fig. 145 this is effected by turning *H*. *H* may, if desired, be motor controlled. After the two machines have been synchronized and are operating in parallel the proper division of the load between them is accomplished by adjusting their governors and adjusting the field rheostats to minimize the cross currents. Division of load cannot be accomplished with only the field rheostats; see the author's American Electricians' Handbook. The machine which is to pull most of the load must be given proportionally more steam.

152. The Care Of Governors seldom includes anything more than oiling and occasionally re-packing a stuffing box or regrinding a valve. The operation of the governor should be examined frequently. On small turbines, the whole governor may be moved by hand to see that it moves freely and shuts off the steam. If undue lost motion develops, or if any part of the mechanism shows undue friction, the difficulty should be promptly remedied as explained in Sec. 132. There is some simple method of making a small change in speed on nearly all governors; and sometimes adjustable weights are provided to change the regulation as in Sec. 147. But the manufacturer should be consulted before any extensive or radical adjustments are made. After any governor adjustment, the action of the device throughout its range should be noted to make sure that it is safe.

Note.—The Elaborate Relay Governing Mechanisms Employed On Large Turbines Are Too Involved And Various To Admit Of Special Directions For The Care Of All Of Them. In general,

there should be means of ascertaining at all times if the relay system is properly filled at the proper pressure with the operating fluid (usually oil); see Div. 10. There is, usually on large turbines, an emergency oil pump (Sec. 197) which will keep up the pressure in the oil system if the regular pump becomes inoperative. The governor proper of a relay governor operates exactly as do other spring-loaded fly-ball governors.

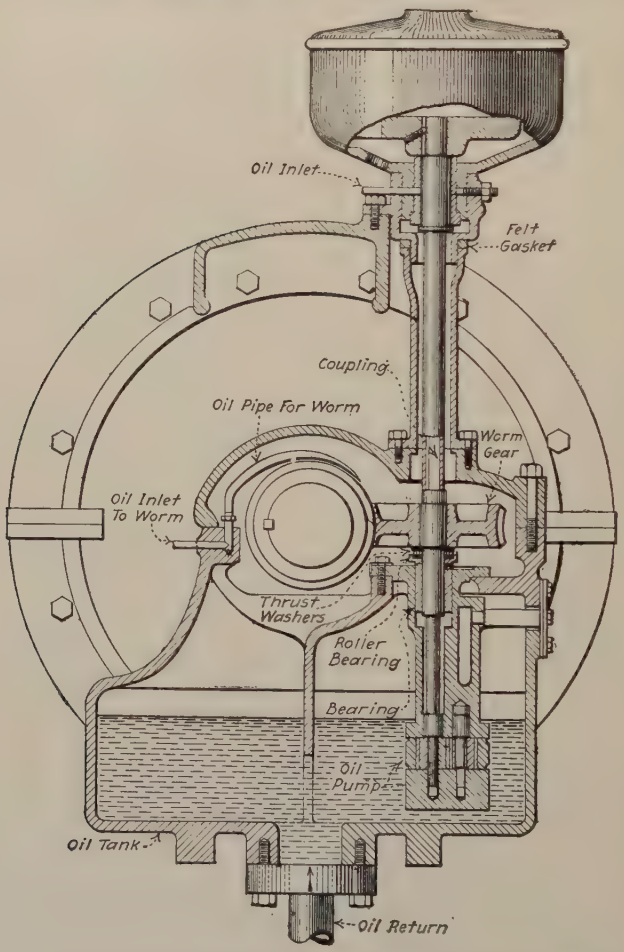


Fig. 161.—Illustrating the lubrication of a *General Electric Co.* Curtis turbine governor.

153. The Emergency Governor Should, Preferably, Be Tested Daily by carefully overspeeding the turbine up to the

speed at which the emergency governor should operate. When thus testing, the speed, as indicated by a tachometer, should be carefully watched. It should never be assumed that the emergency governor is unnecessary because the speed governor functions properly. Additional protection against overspeed is needed.

NOTE.—THE PARTS OF A STEAM TURBINE GOVERNOR WHICH REQUIRE THE MOST OILING are the worm gears and thrust bearings. These are sometimes provided with forced-feed oil systems as in Fig. 161; see also Div. 10. It is very important that the linkage pivots be kept oiled and not be allowed to stick but as these move but little, they do not require much oil.

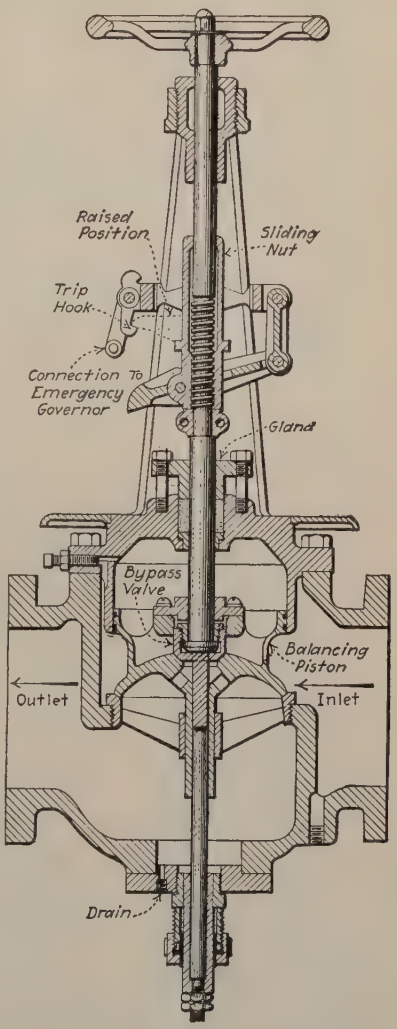


FIG. 162.—Throttle valve with safety-stop attachment used on some *General Electric Co.* Curtis turbines. (Many are in use but they are now applied to new machines only in special cases.)

154. The Principal Kinds Of Valves Used In Connection With Steam Turbines are: (1) *Throttle valves* (Fig. 162) which are used for admitting steam by hand to the turbine. (2) *Safety-stop or emergency valves* (Fig. 137) which are operated by the emergency governor, sometimes the emergency governor trips the throttle valve. (3) *Governor valves* (Figs. 129 and 152) which are operated by

the speed governor. (4) *Nozzle valves* (Fig. 163) which are used principally on small turbines for admitting steam to additional nozzles for heavy loads. (5) *Bypass or stage valves*,

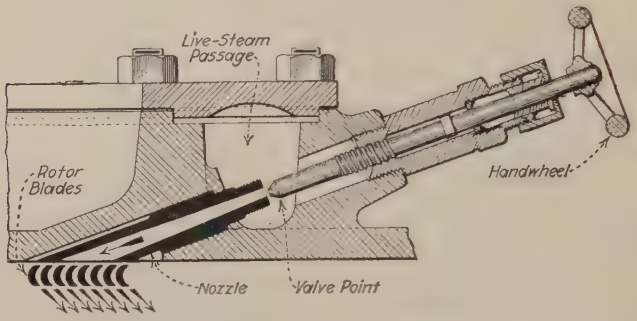


Fig. 163.—*De Laval* nozzle and valve.

(V_2, Fig. 78) which are used for admitting steam to later stages of a multi-stage turbine to carry overloads; these valves may be operated by hand or by the speed governor. (6) *Relief valves* (Fig. 164 and Fig. 29) which are safety valves placed in the turbine casing to protect it against excessive pressures. (7) *Atmospheric relief valves* (Fig. 185) which allow the turbine to exhaust to the atmosphere if the condenser fails and thereby prevent the building up of an excessive pressure in the turbine casing; such valves are connected as side outlets in the exhaust pipe between the turbine and the condenser. See the author's Steam Power Plant Auxiliaries And Accessories.

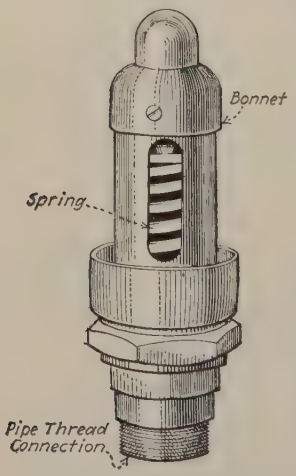


Fig. 164.—A relief valve suitable for use on a steam turbine. (*Ashton Valve Co.*)

Note.—Throttle Valves For Small Turbines are usually ordinary globe valves in the steam pipe near the turbine. For larger turbines, the throttle valves are more elaborate as shown in Figs. 151 and 162, and act also as safety-stop valves. The

balancing pistons of these valves are subject to some of the troubles of engine pistons, although a certain amount of leakage past these pistons is expected.

Note.—A Sentinel Valve (*Kerr Turbine Co.*) is a valve which is so placed and designed as to allow escape of steam and thereby give warning if the pressure becomes high in the low-pressure end of the turbine casing. *Overload valves* are valves which are opened to carry overloads, that is to give the turbine more power than its normal rating. They are, ordinarily, stage valves or nozzle valves and may be operated either by hand (for small turbines, usually) or by the speed governor (for large turbines) depending on the construction employed.

155. The Chief Troubles With Valves Are: (1) *Stuffing-box leaks;* (2) *Valve leaks or breaks;* (3) *Sticking.* Stuffing-boxes can be repacked with various types of high-temperature packings which are on the market for the purpose. For most saturated-steam valve stems, candle-wicking soaked in oil may be used. A governor-valve stem must be packed very carefully so that it will hold steam without much friction of the packing. It is usually better to first screw the gland nut up tightly and then slack it off so as to relieve the pressure on the stem. In general, it is better to have a slight steam leak around a governor-valve stem than to have too much friction. Some indications of a leaky governor valve are: (1) *Racing at light loads with the valve apparently closed* and (2) *heating of the governor thrust bearing due to the force developed by the governor in endeavoring to close a leaky valve.* One test for valve tightness is to close the valve by hand while the turbine is running and note how rapidly its speed decreases.

Note.—Common Causes Of Governor Valve Failure are wet steam and running constantly at light loads. Wet steam may be avoided by lagging the steam pipes and installing a separator. Running at light loads will not wear the valve if one or more of the nozzle valves are turned off. If this cannot be done, a smaller valve should be used. It is necessary to ascertain from the manufacturer what is the smallest valve which will carry the required load. If a conical-seated valve is reground occasionally, it may be kept in good condition in spite of continued running at light loads.

156. Steam Strainers (Figs. 152 and 165) are provided in the admission passages of most steam turbines. They are usually located so that the steam is strained before it passes the gover-

nor valve. This is a precaution to prevent particles of scale from the pipe and other foreign matter from getting under the governor valve and preventing its shutting. Strainers are commonly constructed of sheet metal in which holes are punched which are sufficiently large to allow the necessary flow of steam but small enough to keep out any solid particle which would damage the turbine. The total area of the holes is

FIG. 165.—Common type of steam strainer used on small turbines.

generally made much larger than that of the rest of the preceding and following passages so that there will not be much friction in the strainer.

QUESTIONS ON DIVISION 6

1. Under what conditions may a turbine be operated without a governor? Why is a governor usually necessary?

2. Show by a sketch the action of an elementary direct throttling governor.

3. Of what principal parts does a complete governing mechanism for a large turbine ordinarily consist?

4. In what three ways may steam turbine governors be classified? Name at least two subclasses under each classification.

5. What is one disadvantage of a throttling governor? One advantage?

6. Explain the use of a block or key in adjusting the lost motion on a Sturtevant governor.

7. What is the function of a vacuum breaker on a governor?

8. What method may be used for making speed adjustments of about 2 or 3 per cent. on nearly all horizontal throttling governors? What should be done in case it is desired to make a radical speed adjustment on a governor?

9. How may corrosion of governor valves be minimized?

10. What is an emergency governor? Show by a sketch how a simple one may function.

11. At about how much greater than normal speed is the emergency governor usually set?

12. What is the function of an oil-relay mechanism for a steam-turbine governor? Draw a sketch of and explain the operation of such a mechanism.

13. What is the advantage of an intermittent governor over a throttling governor?

14. Explain, using a sketch, the action of a floating lever in a relay governor.

15. What is the effect of decreasing the number of coils of a governor main spring?

16. How does the Westinghouse safety stop control the automatic throttle valve? Use a sketch in explaining.

17. What is the purpose of bypassing in a multi-stage turbine? What are its disadvantages?

18. How do multi-ported governor valves avoid loss of energy by throttling?

19. What is the function of an inertia arm in a governor?

20. Explain the operation of a pneumatic turbine governor. What are its advantages?

21. To what steam engine mechanism may the Rice mechanical valve gear be compared? Explain the Rice governor using a sketch.

22. What is the function of a dash-pot on a governor? How may the piston on one be made to move more slowly? More rapidly?

23. How are turbo-alternators usually synchronized from the switchboard?

24. What is a throttle valve? How may it be interconnected with an emergency governor? Explain with a sketch.

25. What are bypass valves? Atmospheric relief valves?

26. What is a sentinel valve? A relay valve?

27. What are the three chief troubles encountered in valves?

28. How may leakage in a governor valve be detected? How repaired?

29. What steam and load conditions tend to wear out the valves of throttling governors?

30. What is the general construction of most steam strainers for turbines? What is their function?

DIVISION 7

STEAM-TURBINE REDUCTION GEARS AND COUPLINGS

157. The Function Of A Steam-turbine Reduction Gear (Fig. 166) is solely to "reduce" the rotative speed of the turbine shaft to a suitable speed for driving some other machine.

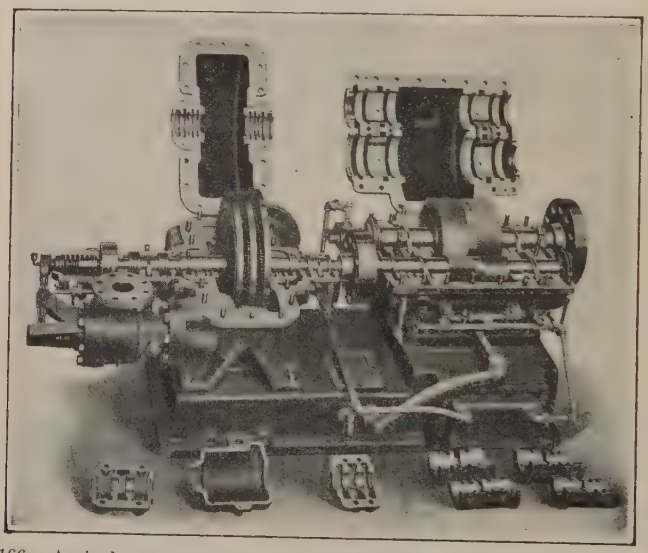


FIG. 166.—A single-stage *Moore* steam turbine, showing the method of mounting turbine and reduction gears on a common bedplate.

Since turbines can be operated efficiently only at high rotative speeds (see Div. 3) and since many mechanically driven machines must be operated at low rotative speeds, it is obvious that these low-speed driven machines cannot be coupled directly to the turbines. Strictly speaking, a reduction gear does not reduce the speed of the turbine shaft. Rather, the turbine shaft transmits its power through the

reduction gear (or gears) to another shaft which then is connected to the driven machines.

Note.—Reduction Gears Are Often Not Necessary With the following machines: (1) *Alternating-current generators.* (2) *Small direct-current generators* (below about 50 kw.). (3) *Centrifugal pumps.* (4) *Fan blowers.* (5) *Turbo-compressors.* Nearly all other machines must be driven at much lower speeds than those at which steam turbines operate and, hence, require reduction gears.

158. Steam-turbine Reduction Gears May Be Classified as follows: (1) *Single-reduction gears,* Fig. 166. (2) *Double-*

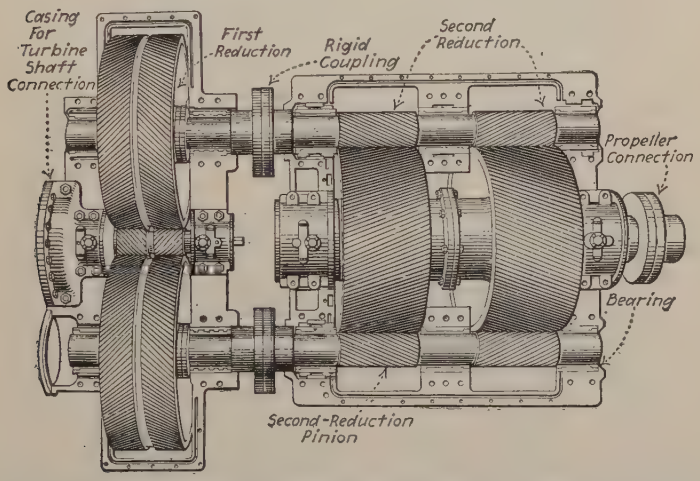


Fig. 167.—Single-plane-type, double-reduction gears for a 3,000-hp. marine turbine which reduces the speed from 3,500 r.p.m. at the turbine to 90 r.p.m. at the propeller. (*De Laval.*)

reduction gears, Fig. 167. (3) *Epicyclic gears,* Fig. 171. Single-reduction gears may be employed whenever the turbine speed does not exceed about six or eight times the speed of the driven machine. Double-reduction gears are employed for greater speed reductions than can be accomplished with a single reduction. By employing a double reduction the sizes of the gears may be kept smaller than if the total reduction were accomplished with one gear and one pinion. The epicyclic-gear speed reducer is explained and discussed in Sec. 162.

Note.—Double-reduction Gears Are Used Extensively With Marine Turbines and occasionally for such stationary service as mill or shaft drives. Double-reduction gears whose shafts all lie in one plane

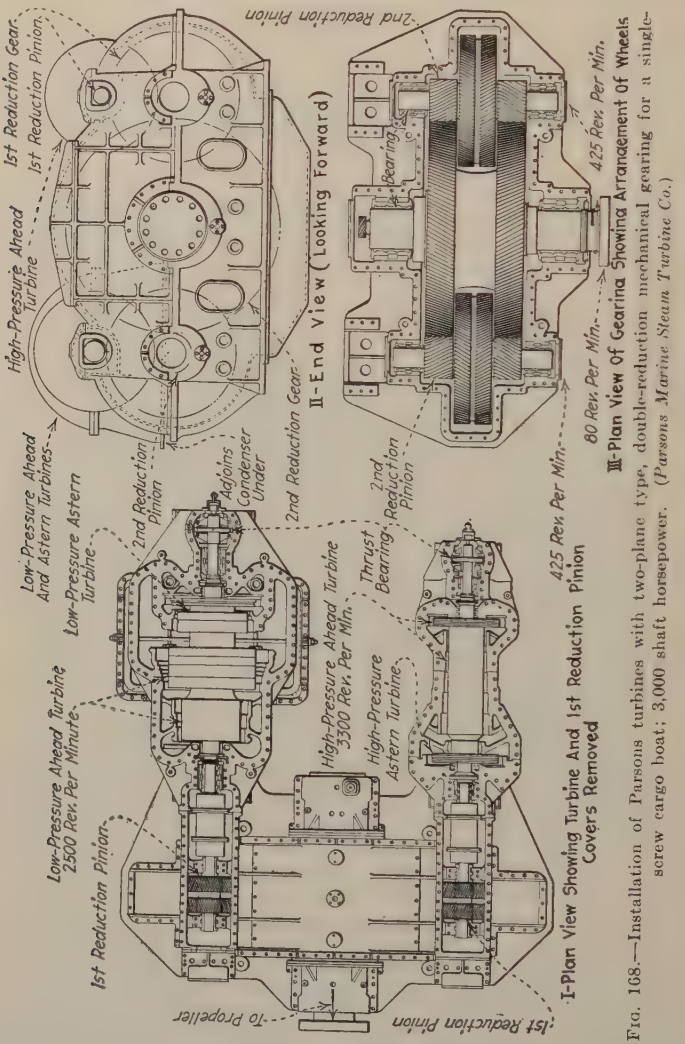


Fig. 168.—Installation of Parsons turbines with two-plane type, double-reduction mechanical gearing for a single-screw cargo boat; 3,000 shaft horsepower. (*Parsons Marine Steam Turbine Co.*)

(Fig. 167) are called *single-plane gears* to distinguish them from those in which the driven shaft is located lower than the turbine shaft. *Two-plane gears* (Fig. 168) usually transmit power from two turbines to a single slow-speed shaft.

159. The Construction Of Reduction Gears is usually such that the gears are enclosed in a case (Fig. 169) which serves to exclude dust and other foreign matter from the teeth. The gears are usually cut from high-grade rolled steel. The teeth are of the double-helical or herringbone type and thus

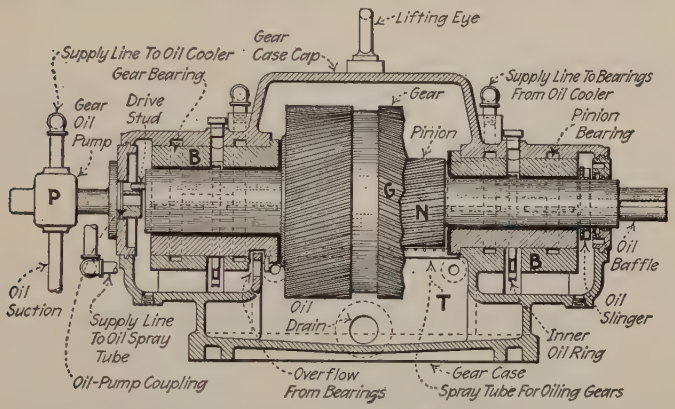


FIG. 169.—Side sectional view of double helical reduction gears. (*Moore Steam Turbine Corporation*, Wellsville, New York. Instruction Card No. 4.) *Forced-feed lubrication* is used in all Moore reduction-gear sets. The oil is supplied from a geared pump, *P*, under pressure, to the bearings, *B*, and also is sprayed through small holes in a copper pipe, *T*, onto the gear, *G*, and pinion, *N*, at the pitch line. Stop cocks are provided in the feed lines to the bearings for regulating the flow and also in the supply line for spraying oil onto the gears. These cocks should be adjusted so both bearings and gears will receive a liberal supply of oil. The bearings should be given all they will take without overflowing.

Inspection of gear lubrication can be made through the opening (not shown) which is provided for this purpose. A metallic ringing sound is an indication that the gears are not getting sufficient oil. If for any reason too much oil is fed to the bearings and gears, so that it is not carried away fast enough through the drain pipe and that it backs up in the case until the gear dips in the oil, there will result undue heating, caused by the oil being thrown against the sides of the case. The remedy is to reduce the quantity of oil which is used.

A cooling device is provided in the form of a *brass-tube cooler* or *plate-type cooler* for cooling the oil. Water is used for cooling. The oil is circulated from the discharge of the oil pump through the brass-tube cooler. In the plate cooler, the oil passes over the cooling surface when it is being returned to the suction tank.

provide smooth quiet operation which is free from vibration and end thrust. The gears are supplied with oil, as are also the bearings, from a pump (see Div. 10) which is driven from the end of the large-gear shaft. The oil is cooled by passing it over a water-cooled coil or plate and is then returned to the pump. Some large marine reduction gears are so designed

that the pinion shaft turns in a *floating frame* carried on hydraulic rams. This elastic support of the pinion shaft renders the gearing practically noiseless and insures automatically more nearly perfect alignment between gear and pinion under all conditions. With turbines of smaller output, however, the floating frame is seldom used.

Note.—The Transmission Efficiency Of Reduction Gearing is very high; it may exceed 98 per cent. The *transmission efficiency* = (*the power delivered at the low-speed shaft*) ÷ (*the power developed by the turbine*). This efficiency is materially decreased, however, if the oil level is permitted to reach such a height that the gear dips into it or if too little oil is supplied to the gears.

160. Troubles With Reduction Gears are infrequent. The principal care which reduction gears require is to see that they are maintained in proper alignment and that they are properly lubricated. *Misalignment* causes vibration and rapid wear and is frequently the cause of noisy operation. When lining up the gears bear in mind that either the gear or the pinion, depending on the direction of rotation, will be lifted to the top of its bearings when the gears operate. When the gears run toward each other at the top the pinion will lift. When the gears run away from each other at the top the gear will lift. Note the clearance in the bearings by lifting on the shaft; the clearance is the amount of "give" of the shaft in the bearings. Then make adjustment for about 0.002 in. less than the observed clearance. For the *lubrication* of high-speed reduction gears a good gear oil should be used. See Div. 10. The oil should be kept clean by renewing or filtering it as often as is found necessary. The temperature of the oil should be maintained at between 130° and 180° F.

Note.—The Oil-cooling Coils Of Reduction Gears should be supplied with cool clean water in sufficient quantity that the oil is kept at the proper temperature (see above). The water piping should be arranged that the coils may be protected against possible freezing.

161. The Alignment Of Reduction Gear, as given by the *Westinghouse Electric and Mfg. Co.* in "Instruction Book No. 5,220" is as follows:

1. ALIGNMENT IN A HORIZONTAL PLANE.—Check the alignment with the block gages furnished. If these are unavailable, caliper between the aligning collars (*C*, Fig. 170) on pinion and gear wheel and note the micrometer measurement. Micrometer the gear and the pinion-aligning collars. The center to center distance between gear wheel and pinion shafts is thus determined by calculation from these measurements on each side of the gear wheel, and should of course be the same, within 0.001 in. If it is not, shift a liner from the proper side pad of the pinion bearing, *B*, to the opposite side. A 0.000,5-in. liner will affect the difference in center-distance dimensions about 0.001 in. Adjust at the pinion bearing, *B*, in preference to the turbine bearing, *A*, since the latter throws the glands slightly more out of center.

2. ALIGNMENT IN THE VERTICAL PLANE.—This alignment can be

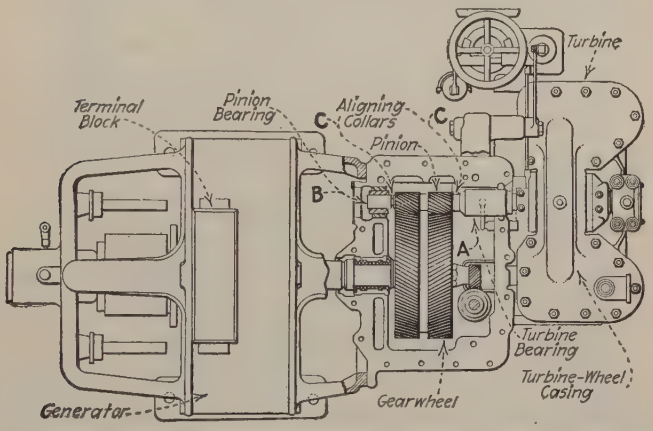


FIG. 170.—Small (15–50 kw.) geared-turbine and generator. (*Westinghouse Electric & Mfg. Co.*)

properly checked only by the operation of the unit. As a rough approximation, coat a few pinion teeth with Prussian blue and pull the turbine rotor around in the direction of its rotation. Then note the distribution of the contact marks on the gear teeth. If these seem to be concentrated at the ends of the teeth, say at the turbine end of each helix, raise the pinion bearing, *B*, by shifting a liner from the top pad to the bottom one and repeat till the contact appears distributed rather than concentrated. This is not a complete check, since, under load, the pinion takes a slight deflection. To thoroughly check, prepare the gear-wheel teeth by washing them with a copper-sulphate solution, thus giving a light film of copper deposit which will plainly show the contact of the teeth during operation. When everything else about the unit is ready, run the turbine for half an hour under approximately full load. Then remove the gear case cover and examine the contact marks on the gear teeth.

These should extend from end to end of the teeth. If the marks are concentrated at either end, transfer a 0.005-in. liner as directed above, again apply the copper-sulphate solution and repeat the trial run. Closer pad adjustment than 0.005 in. is not required, even though the tooth contact marks might seem to indicate it.

3. Backlash Or Clearance Of Teeth.—Block the gearwheel against end movement. Push rotor and pinion to one end as far as possible and take a feeler-gage measurement where convenient, say

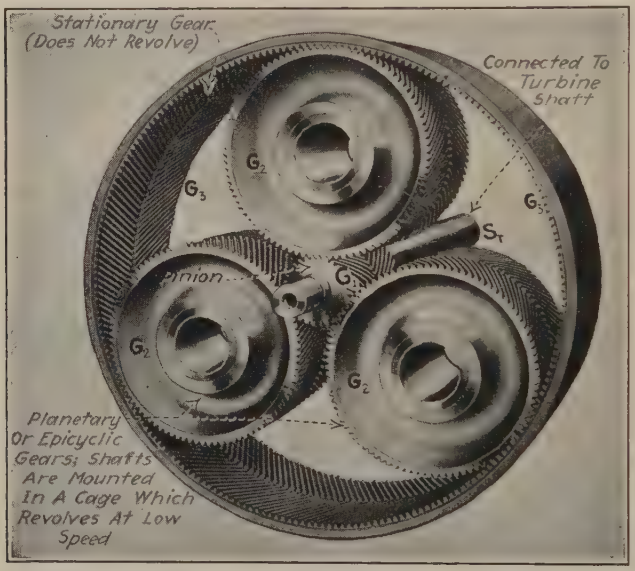


Fig. 171.—Illustrating the principle of the "Turbo-Gear" speed reducer: Annular gear G_3 is so held in the frame of the unit that it cannot revolve; pinion shaft S_T is revolved at high speed. (Epicyclic reducing gears as manufactured by the *Poole Engineering and Machine Company.*)

between the gland runner and casing. Pull the rotor in the opposite direction and again take a feeler measurement. The difference, or end play of the pinion should be between 0.009 and 0.016 in. In taking such measurements be sure that glands or blades do not strike adjacent parts, thus giving false values. If necessary to correct end play, alter center distance by shifting equivalent liners of both turbine and pinion bearings, *A* and *B*, from one side to the other. Operation (1) has already put the shafts parallel and therefore one bearing should not be changed without changing the other the same amount. The end play will be changed about 0.005 in. by shifting a 0.005 liner.

162. Epicyclic Reducing Gears (Fig. 171) are so formed that, although they afford but a single reduction, the driven

or low-speed shaft has its axis exactly in line with the driving or high-speed shaft. They are installed in a frame (Fig. 172) which presents the same general appearance as an enclosed electric motor or generator. Under certain conditions their construction makes them more applicable than ordinary single-reduction gears. Their operation is obvious from Fig. 171.

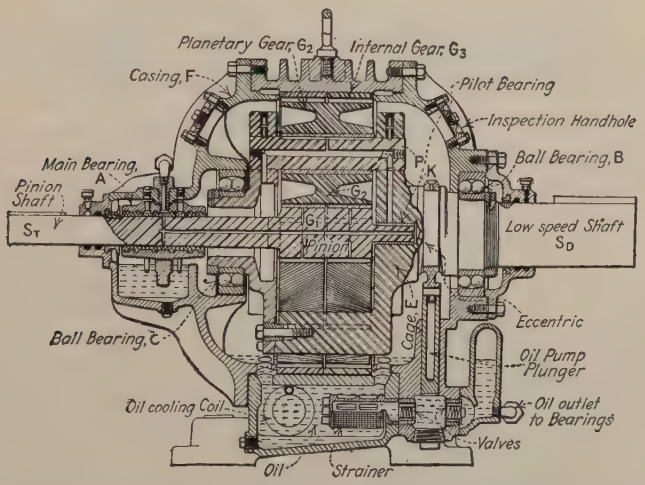


Fig. 172.—Longitudinal section through the *Turbo-gear* speed reducer, Fig. 171. The low-speed shaft, S_D, carries the cage, E, and is supported in the casing, F, on the two ball bearings B and C. The cage, E, contains 3 pins, P, upon which the planetary gears G_2 revolve. The pinion shaft, S_T, carries the pinion, G_1, which meshes with the three planetary gears, G_2; S_T is carried in the two bearings A and K. The planetary gears G_2 "roll around" in the internal gear G_3, which is held stationary—so that it cannot turn—in the casing, F. An oil pump is driven by the eccentric on the low-speed shaft, S_D.

163. Steam-turbine Couplings Are Of Two Kinds: (1) *Rigid*, (Fig. 173) (2) *Flexible* (Fig. 174) see Sec. 164. Rigid couplings are employed principally on small turbines and only where both the coupled turbine and driven shaft are supported on only two or three bearings. Where four bearings are used, two for the turbine shaft and two for the driven shaft, a flexible coupling (Sec. 165) is always employed.

Note.—The Rigid-coupling Two-bearing Unit Is Very Desirable for small-power machines. There is much less chance of such a machine

getting out of alignment and thus giving bearing trouble. A two-bearing unit also occupied less floor space than does a three- or four-bearing unit. The *Steam Motors Co.* of Springfield, Mass. specializes in two-bearing units which it builds in sizes up to 300 hp.

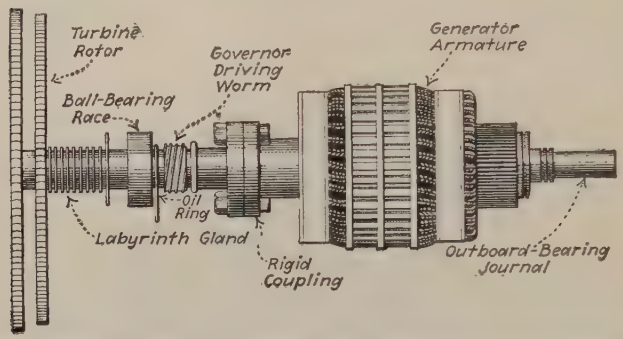


Fig. 173.—Assembled rotor for a "Steam Motor" generator-set showing rigid flange coupling. (*The Steam Motors Co.*)

164. The Purpose Of Flexible Couplings In Steam-turbine Drives is: (1) *To provide for any slight inequality in the wear of the bearings.* (2) *To permit axial adjustment of the turbine spindle.* (3) *To allow for differences in expansion.* It is obvious that two shafts, each supported on two bearings, would be bent by any deviation of their bearings from one straight line. Furthermore, it is very difficult to exactly align four bearings into a straight line and, if aligned, to so maintain them. Hence, and to permit of axial adjustment of the two coupled shafts, a so-called flexible coupling (Fig. 175) is employed; see Sec. 165.

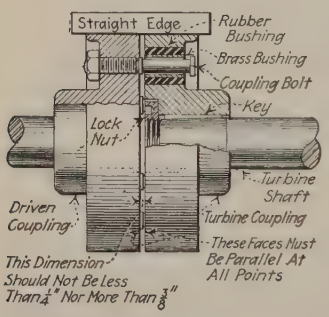


Fig. 174.—Final alignment of Type-6 *Sturtevant* turbine coupling. The turbine rises when steam is turned on. Therefore provide allowances to compensate for this change. It is important that final alignment be made under operating temperatures.

165. There Are Three Principal Types Of Flexible Couplings, namely: (1) *Rubber-bushing type*, (Fig. 174), wherein a number of —usually six—coupling bolts or pins are fastened rigidly to

one half of the coupling and are extended through rubber (or leather) bushings in the other coupling half. The rubber affords the flexibility. (2) *Flexible-pin type*, (Fig. 175) wherein flexibility is attained through the bending of small driving pins

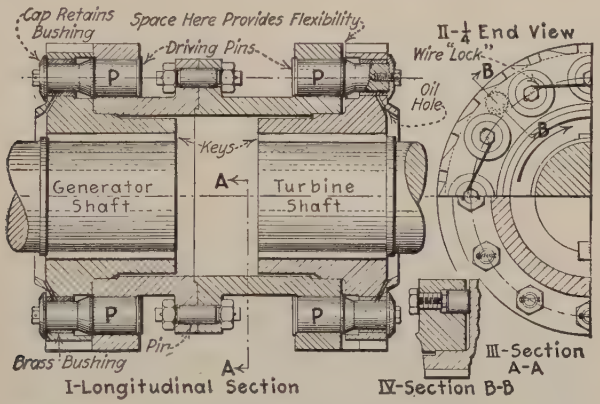


Fig. 175.—Pin-type flexible coupling used on *Westinghouse* turbines.

P—this type employs no highly compressible material; sometimes the pins, *P*, are built up of small sheet-steel laminations. (3) *Claw* or *jaw type*, (Fig. 176) wherein flexibility is attained

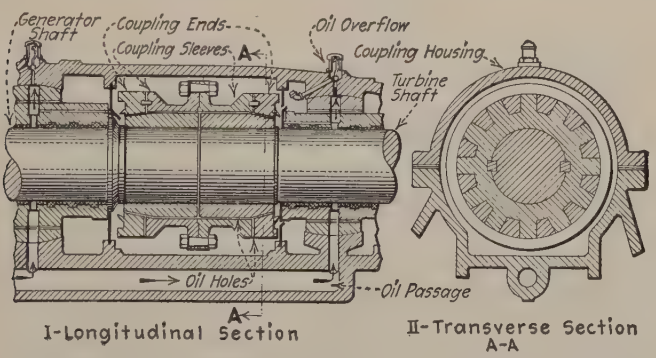


Fig. 176.—Claw-type flexible coupling used on *Allis-Chalmers* turbines.

through the joints between the coupling jaws and the claws on the sleeves. Couplings of types (2) and (3) require lubrication of the driving surfaces because there is sure to be some sliding between the metal contact parts.

Note.—The "Flexibility" Of A Flexible Coupling is very small; that is, a flexible coupling will permit of very little misalignment of the two shafts which it connects. Under operating conditions (turbine hot) there should not be over 0.002 in. difference in height between the two halves, nor should the angular misalignment between the connected shafts be such that the difference in opening between the two halves on opposite sides of the shaft exceeds 0.002 in.; (see Fig. 174) and Sec. 167. The principal mode in which a flexible coupling affords much "play" is in the axial direction.

166. The Care Of Steam-turbine Couplings is simple. Rigid couplings, once installed, require no further care. The bolts must be so fastened, however, that they cannot come out—note the "wire-lock" fastenings in Fig. 175. All-metal flexible couplings must always be lubricated. All flexible couplings should be examined periodically (say once a month) to see that the connected shafts have not become misaligned by wear or other causes. Should the couplings need aligning proceed as directed in Sec. 167. Coupling parts which, when an inspection is made, show considerable wear should be repaired or the worn parts replaced.

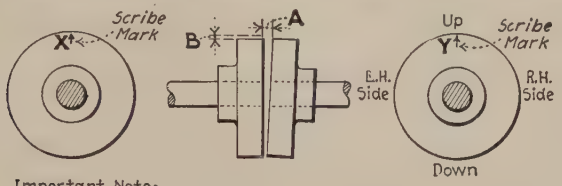
Note.—Serious Misalignment Of Shafts Results In *vibration, burned-out bearings, broken shafts, broken couplings,* or *broken other rotating parts.*

167. A Convenient Method Of Aligning Two Shafts At Their Coupling is given below. Two shafts may suffer from two kinds of misalignment. They may be out of line sideways (the ends of their axes not meeting) or they may be nonparallel. The following method of checking their alignment is simple, always applicable, and can be performed in a few minutes:

Explanation.—With a pin-type coupling, insert a coupling pin, without its bushing, through both halves of the coupling and leave this in while measuring. *During all of the following measurements see that the couplings are pushed as far apart as the thrust bearings will permit.* Make two marks, X and Y, one on each coupling, as shown in Fig. 177. With these points up, as shown in Fig. 177, measure distance A using a feeler or thickness gage. Measure also distance B using a steel straightedge, as shown in Fig. 174, and a feeler gage. Record these distances as shown in Fig. 177. Then turn the points to the right-hand side and repeat the measurements at the marked points. Repeat the measurements with the points in the down- and left-hand positions. If all of the dimensions

A are the same, the two shafts are parallel. If all of the measurements *B* are the same, then the two shafts are not out of line sideways. If both of these conditions are not fulfilled, the shafts should be adjusted by shifting or shimming the bearing pedestals or linings until the shafts are perfectly aligned.

With a claw-type coupling, a test rod, *C* (Fig. 178) should be clamped



Important Note:
The Two Shafts Must Always Be Turned Over Together While Measuring So That Points **X** *And* **Y** *On Each Of The Couplings Are Always Opposite Each Other*

STANDARD TABLE OF DIMENSIONS TO BE OBTAINED EACH TIME ALIGNMENT BETWEEN TWO SHAFTS IS CHECKED

Distance "A" At Points "X" And "Y" As Shafts Are Turned Over Together To Varying Positions		Distance "B" At Points "X" And "Y" As Shafts Are Turned Over Together To Varying Positions	
Position	Inches	Position	Inches
Up	0.124	Up	0.008
R.H. Side	0.124	R.H. Side	0.008
Down	0.124	Down	0.008
L.H. Side	0.124	L.H. Side	0.008

The Above Indicates That Shafts Are Parallel And In Line. If Dimensions Are Not Constant For Every Position, Then They Should Be Made So By Shifting Or Shimming Under Feet Of One Of The Members.

FIG. 177.—Example, illustrating method of aligning couplings.

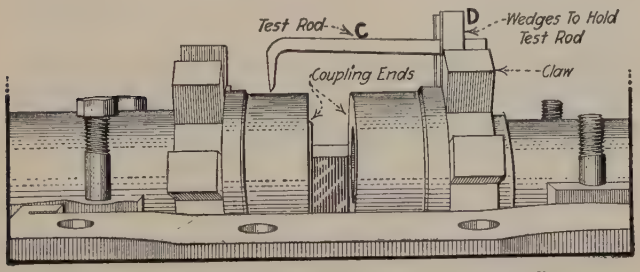


FIG. 178.—Showing a method of aligning a claw coupling.

in one of the coupling ends as at *D*. The distance between the other coupling-end and the point of *C* should be measured with a feeler gage as was distance *B*, Fig. 177. The distance between two claws directly opposite each other should be measured (with calipers or micrometer) in the same manner as was distance *A*, Fig. 177. For the shafts to be in line, these two distances must be the same for any position—whether the points measured are up, down, or on the right or left side.

QUESTIONS ON DIVISION 7

1. Explain the function and purpose of reduction gears.

2. What classes of machines are frequently driven without reduction gears? Name some with which gears are necessary.

3. Name and distinguish between the three principal types of reduction gears.

4. What are the principal uses of single-reduction gears and give their limitations.

5. What are the principal uses of double-reduction gears and what two types are there? What determines largely which type is required?

6. Explain, with a sketch, the usual construction of reduction gears. Describe the *floating frame* construction.

7. Define *transmission* efficiency. What is a usual value and what may lower it?

8. Explain what care reduction gears require and what troubles must be guarded against.

9. Describe fully, using sketches, the method of aligning the teeth of a pair of reduction gears.

10. Explain, with a sketch, the operation of epicyclic reduction gears. What are their advantages?

11. In general, what two types of couplings are employed on steam turbines?

12. On what kinds of machines are rigid couplings employed? What are the advantages of such drives?

13. Give three reasons for employing flexible couplings.

14. Describe, using sketches, the three principal types of flexible couplings. Which types require lubrication?

15. What can you say regarding the "flexibility" of the so-called flexible couplings?

16. What care do steam-turbine couplings require, if any?

17. What harmful results are occasioned by poorly aligned turbine shafts?

18. Explain, with sketches, methods of aligning pin and claw couplings.

DIVISION 8

STEAM-TURBINE REGENERATORS AND CONDENSERS

168. A Steam-turbine Regenerator Or Accumulator (Fig. 179) consists of a large mass of water, *W*, which absorbs heat from exhaust steam when the steam is brought to it and which gives up heat by evaporation when required. A regenerator is

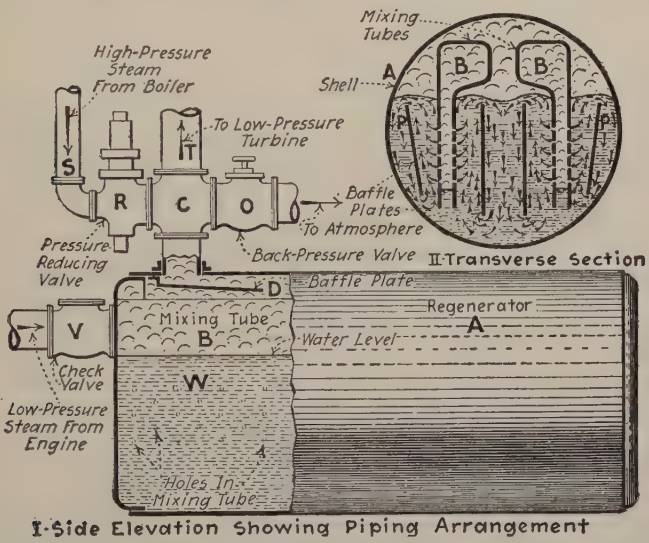


FIG. 179.—A typical Rateau regenerator or accumulator for use with low-pressure turbines.

generally necessary when exhaust steam from an intermittently used non-condensing steam engine, such as a rolling-mill engine or a steam hammer, is used to drive a low-pressure turbine (Div. 9). A regenerator will insure a steady flow of steam to the turbine for a short time (about four minutes, usually) after the engine has been stopped. Regenerators should always be enclosed in an effective heat-insulating jacket.

Explanation.—The regenerator of Fig. 179 consists of a shell, *A*, which is kept about two-thirds full of water and which contains two mixing tubes, *B*. Exhaust steam from the engine is led, first through an oil separator (not shown), and then through a check valve, *V*, into tubes, *B*. A slight steam pressure in *B* forces the water down in their vertical legs which have a large number of holes (usually about ¾ in. in diameter) as shown. The steam then issues through these holes and bubbles upward or condenses, depending upon the temperature of the water, causing a circulation as shown by the arrows in *II*. The circulation is assisted by baffle plates, *P*. As soon as the water in, *A* reaches its boiling point, the space above the water level will fill with steam which then passes outward through cross, *C*, and pipe, *T*, to the low-pressure turbine. The baffle plate, *D*, prevents small drops of water from passing out through *C*.

If, now, the turbine does not require as much low-pressure steam as is furnished by the engine exhaust, the steam will not be permitted by the turbine governor to flow through *T*. Hence, it will accumulate in *A* and raise the pressure. But, as the pressure in *A* increases, the boiling point of the water also increases. Hence the water will now absorb more heat. Thus more and more heat is stored in the water until, finally, the pressure in *A* reaches a value at which the back-pressure valve *O* is set to open. Then all steam which is not needed by the turbine will be discharged into the atmosphere.

If, now, the engine should be stopped, *O* will close and the turbine will draw steam from *A*. Thus the pressure in *A* will be gradually decreased. But, as the pressure is decreased, the boiling point of the water in *A* will be lowered and some of the water will be evaporated. Thus, steam will continue to be supplied to the turbine, but at a gradually decreasing pressure, until the engine is again started. Sometimes, however, a high-pressure steam pipe, *S*, is arranged with a reducing valve, *R*, to admit steam to *C* when the pressure in *A* falls below a predetermined value for which *R* is set. If the turbine is equipped with a bypass or high-pressure valve, the reducing valve, *R*, is not necessary at the regenerator.

Note.—Piping Accessories Which Should Be Installed With A Regenerator are: (1) *An oil separator;* oil is generally undesirable in a steam turbine because it tends to adhere to the blading and clog the passages. (2) *A check valve, V* Fig. 179, to prevent water from passing from the regenerator back to the engine cylinder when the engine is stopped. (3) *A safety or back-pressure valve, O* Fig. 179, to prevent an excessive pressure in the regenerator which might be destructive to the turbine or the regenerator itself. (4) *A float-valve water-level control*, not shown in Fig. 179, to prevent an excessively high water level in the regenerator; the water level will gradually rise as steam is condensed by the loss of heat from the regenerator shell by radiation. The water level may also rise because of the moisture which is carried into the regenerator with the exhaust steam. The water discharged by the float valve may be led to the hot well or permitted to flow into the sewer, whichever is most feasible.

169. Regenerators Are Practical Only When the non-condensing engine which supplies the exhaust steam has short-period shut-downs. If the usual shutdown period exceeds three or four minutes, it is generally better to use a mixed-pressure turbine (Div. 9) than to attempt to use a regenerator. But in cases where the shutdown period seldom exceeds one or two minutes, a regenerator is very useful.

NOTE.—BOILER-PRESSURE VARIATIONS MAY BE CONDUCIVE TO THE USE OF A REGENERATOR.—When a large reciprocating engine is suddenly stopped, the boiler which supplied the engine continues to produce steam at the same rate as before. The steam pressure immediately increases and very soon the safety valves are blowing off steam. The regenerator

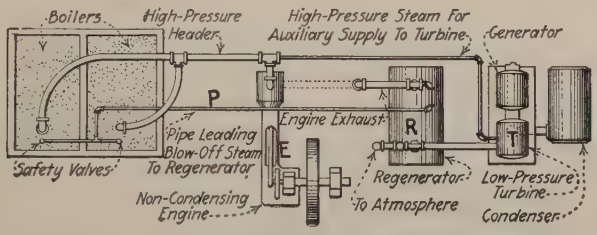


FIG. 180.—Typical layout of a power plant with a non-condensing engine, *E*, regenerator, *R*, and low-pressure turbine, *T*.

can be arranged to receive the steam, which would thus go to waste, in one of two ways: (1) *The blowoff can be piped to the regenerator, R,* **Fig. 180.** (2) *A relief valve may be provided* to discharge steam from the boiler at 1 or 2 lb. per sq. in. less than that for which the safety valves are set, the discharged steam being piped to the regenerator.

170. The Normal Operating Pressure For A Regenerator is generally between atmospheric pressure and 15 lb. per sq. in. gage. A small vacuum could be used but would make difficult the exclusion of air from the system. The relief or back-pressure valve (*O*, Fig. 179) should be set to open at about 2 lb. per sq. in. above the normal operating pressure. The reducing valve or regulator (*R*, Fig. 179) should be set for about 1 lb. per sq. in. below the operating pressure. Hence, the pressure variation in the regenerator should not exceed 3 lb. per sq. in. For economical operation neither the back-pressure valve, *O*, nor the regulator, *R*, should open except when unusual condi-

tions arise. This necessitates the use of a regenerator which contains the proper mass of water (Sec. 171).

Note.—Adjustment Of Regenerator Operating Pressure should be so made, if possible, that the non-condensing engine will supply the same amount of steam as the low-pressure turbine uses. By increasing the regenerator pressure the non-condensing engine can be made to use more steam and the low-pressure turbine less. This, then, is the remedy when the regenerator pressure always is low. By decreasing the regenerator pressure the turbine can be made to use more steam and the non-condensing engine less. Obviously, when the regenerator pressure is always too high (indicated by blowing off), the blowoff valve should be set for a lower pressure.

171. To Compute The Necessary Weight Of Water In A Regenerator, the following formulas may be used. For. (28) gives the weight necessary to insure that the regenerator pressure will not drop too low while the steam supply to it is cut off for a short time. For. (29) gives the weight necessary to insure that a sudden supply to it will not cause a discharge from the back-pressure valve. That formula which gives the greater weight should govern the installation. The formulas are:

$$\mathbf{W}_{W1} = \frac{t\mathbf{W}_{S1}(\mathbf{L}_1 + \mathbf{L}_2)}{2(T_1 - T_2)} \quad \text{(pounds)} \tag{28}$$

$$\mathbf{W}_{W2} = \frac{\mathbf{W}_{S2}(\mathbf{L}_1 + \mathbf{L}_2)}{2(T_1 - T_2)} \quad \text{(pounds)} \tag{29}$$

Wherein: $\mathbf{W}_{W1}$ = the weight of water, in pounds, necessary to insure that the pressure will not fall below a predetermined point while the turbine is using steam but no steam is supplied to the regenerator. $\mathbf{W}_{W2}$ = the weight of water, in pounds, to absorb a momentary rush of steam. t = the maximum time, in minutes, during which steam is being taken from the regenerator while no steam is supplied to it. $\mathbf{W}_{S1}$ = the total steam consumption of the turbine in pounds per minute. $\mathbf{W}_{S2}$ = the weight in pounds of a momentary supply of steam which must be absorbed. $\mathbf{L}_1$ and $\mathbf{L}_2$ = the latents heats of steam, in B.t.u. per pound, at the maximum and minimum pressures in the regenerator. T_1 and T_2 = the temperatures, in degrees Fahrenheit, at the maximum and minimum pressures in the regenerator.

Example.—Determine the weight of water to be stored in a regenerator which operates a 1,000-hp. low-pressure turbine for 4 min. while no steam enters the regenerator. The regenerator pressure may vary between 20 and 17 lb. per sq. in. abs. The turbine uses 30 lb. of steam per hp-hr. Solution.—From steam tables, $T_1 = 228°$ F. $T_2 = 219.4°$ F. $L_1 = 960$ B.t.u. per lb. $L_2 = 965.6$ B.t.u. per lb. Hence, by For. (28), $W_{W1} = tW_{S1}(L_1 + L_2)/2(T_1 - T_2) = 4 \times (1{,}000 \times 30 \div 60) \times (960 + 965.6) \div [2 \times (228 - 219.4)] = 223{,}900$ *lb.*

Example.—If the regenerator of the above problem is to absorb 3,000 lb. of exhaust steam during a short period of sudden supply, how much water should it hold? Solution.—By For. (29), $W_{W2} = W_{S2}(L_1 + L_2)/2(T_1 - T_2) = 3{,}000 \times (960 + 965.6) \div [2 \times (228 - 219.4)] = 335{,}850$ *lb.*

172. A Condenser, As Used In Connection With A Steam Turbine, is a vessel into which the exhaust steam from the turbine is led and wherein the steam is condensed into water or "condensate." The purpose in so doing is to create as high a vacuum as possible in the chamber into which the turbine exhausts. The vacuum is formed by causing the steam to come into contact with cold surfaces, give up some of its heat, and thus change from the vapor to the liquid state. The degree of vacuum formed depends on how rapidly heat can be carried away from the steam. The effect of high vacuum is to greatly increase the amount of heat which is liberated by each pound of steam and which may be converted into work by the turbine. See Sec. 10 for methods of computing the liberated heat at various vacua. See also Div. 13 for the effects of vacuum on steam-turbine economy.

Explanation.—The turbine, *T*, (Fig. 181) exhausts steam at *S*. This steam comes immediately into contact with the tubes inside of the condenser, *C*. Cold water is circulated from *E* to *F* through the tubes. Heat is conducted from the steam through the tube walls to the circulating water. Sufficient heat is thus abstracted from the exhaust steam (about 950 B.t.u. per lb.) so that the steam changes to the liquid state and becomes water. The change from steam vapor to liquid water is accompanied by a great decrease in volume (about 20,000 to 1, at an absolute pressure of 2 in. of mercury) and a corresponding reduction in pressure.

Note.—Surface Condensers Are Generally Used With Steam Turbines. A surface-condenser installation is shown diagrammatically in Fig. 181. *Jet condensers* (Figs. 182 and 183), in which the water comes

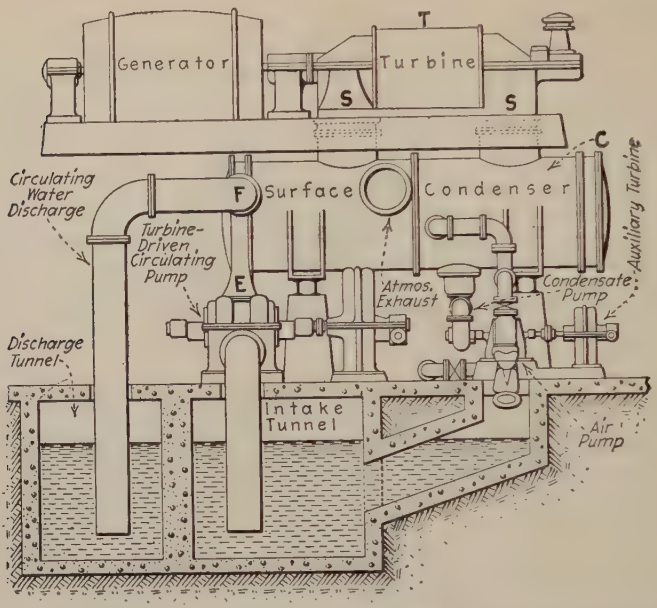


FIG. 181.—Arrangement of equipment in a turbine-driven plant showing surface condenser and auxiliaries.

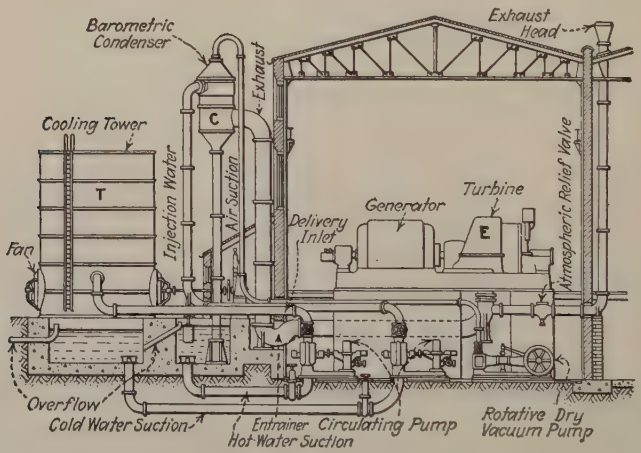


FIG. 182.—Steam-turbine installation with barometric-jet condenser, *C*, and cooling tower, *T*. (*Worthington Pump and Machinery Corp.*)

in direct contact with the steam, may also be used. The surface condenser is better adapted to maintaining a high vacuum than is the jet condenser; also, the surface condenser recovers the feed water in pure form. Therefore, in most cases, the surface condenser is the more econ-

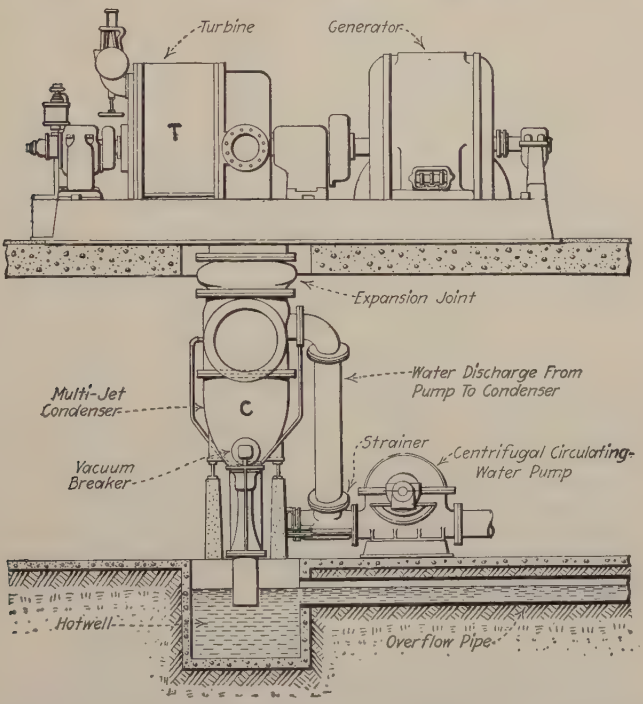


FIG. 183.—Arrangement of a steam turbine, *T*, with a jet condenser, *C*. (*Schutte & Koerting Co.*, "Multi-jet" condenser with which no air pump or condensate pump is required.)

omical for turbine service. For economic comparison between the two types and also for their construction, care, and operation, see the author's STEAM POWER PLANT AUXILIARIES AND ACCESSORIES.

173. To Compute The Necessary Condenser Surface And Cooling-water Requirements For A Steam Turbine, the chart of Fig. 184 may be useful. To use the chart, however, certain assumptions must be made and certain desirable values must be known, as explained below.

EXPLANATION OF USE OF CHART OF FIG. 184.—The average temperature of the cooling-water supply should be first found, either by experi-

ment, from the weather bureau, or by assumption. This determines, to some extent, what vacuum can be profitably maintained. The temperature of saturated steam at the absolute exhaust pressure must be from 25 to 50° F. higher than the cold circulating water—the lower value for high-vacuum work (low absolute pressure, say about 2 in. of mercury) and the higher value for low-vacuum work, say about 4 in. abs. exhaust pressure. In the chart of Fig. 184 the temperatures of exhaust steam are plotted with the absolute pressures along the horizontal axis. The temperature rise of the circulating water should next be computed. The water should not be heated in the condenser to within less than 10°F. of the exhaust-steam temperature. The rate of heat transfer should next be assumed. This may be assumed at 300 B.t.u. per sq. ft. per hr. per degree difference for 4 in. absolute pressure and 350–400 for 2-in. absolute pressure. The use of the chart is illustrated in the following example.

Example.—Assume that it is desired to condense 10,000 lb. of steam per hr. at 2 in. of mercury absolute pressure. Water is available at 70° F. Since steam at 2-in. absolute pressure has a temperature (Fig. 184) of 101° F., the cold circulating water will be 101 − 70 = 31° F. colder than the steam. This (see above) is allowable for a 2-in. pressure. The circulating water may be heated to 101 − 10 = 91° F. Hence, a 20° F. rise in the temperature of the water is permissible. The tubes of the condenser are assumed to transmit 350 B.t.u. per sq. ft. per hr. per degree difference in temperature. What is the necessary capacity of the condenser in square feet? How much water will be required?

Solution.—Find the point *A* (Fig. 184) corresponding to the desired pressure and trace vertically to the 70° F. line at *B*. Then trace horizontally to the 20 degree rise line at *C*. The quantity of water for this rise is 95 gal. per min. for each 1,000 lb. of steam (as read on the diagonal 20° line) or 950 *gal. per min. total* for 10,000 lb. of steam per hr. Now trace vertically to the 350 B.t.u. line at *D*, and thence horizontally to the curve at *E*. The capacity of the condenser may now be read at *F*. The size of the condenser is 127 sq. ft. for each 1,000 lb. of steam per hr. or 1,270 *sq. ft.* for the 10,000 lb. of steam per hr. of this example.

174. In Installing A Condenser To Serve A Turbine, it is customary to locate the condenser below the turbine as shown in Figs. 181 and 183. A short connection between the turbine and condenser serves to minimize the pressure drop between the two and also minimizes the possibility of air leaks. Where space limitations demand it, however, the condenser may be placed on the same floor with the turbine. Figure 182 shows a desirable arrangement of apparatus where a barometric jet condenser, *C*, is used with a steam turbine, *E*, and is supplied by water which is recooled in a tower, *T*. All turbine installa-

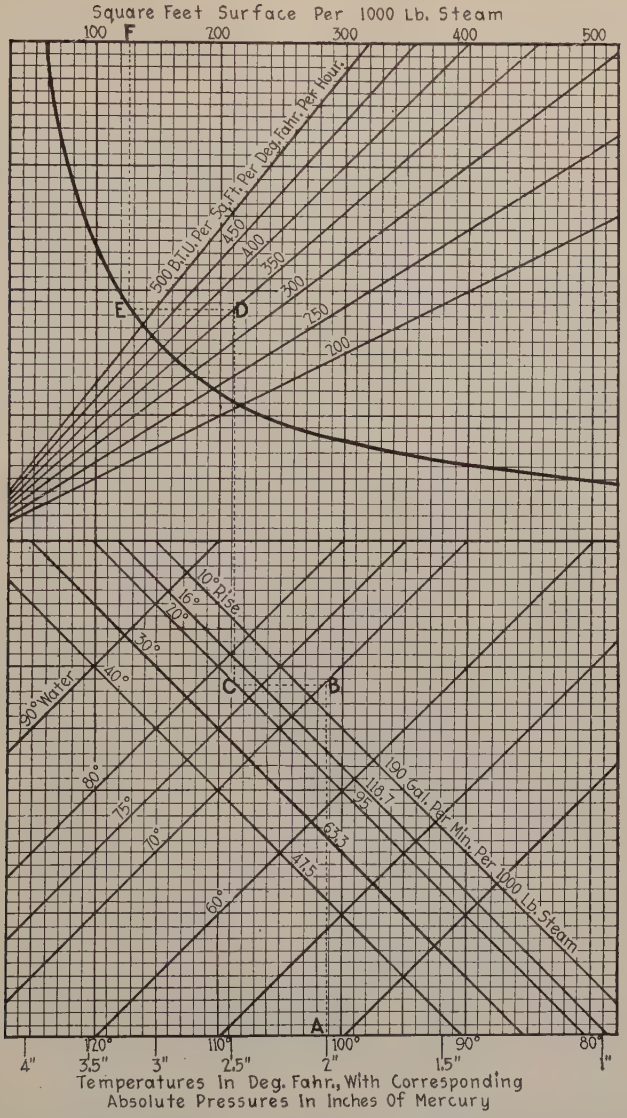


FIG. 184.—Graph, based on a steam rate of 1,000 lb. per hr., for determining the necessary condenser cooling surface and cooling-water volume required under various conditions. (*Worthington Pump and Machinery Corp.*)

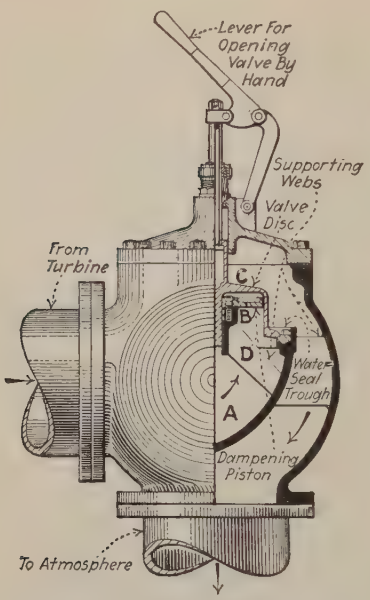


FIG. 185.—Schutte automatic free exhaust (atmospheric relief) valve. Valve disc, *C*, is raised when the pressure in *A* exceeds the pressure of the atmosphere. This pressure is transmitted through the small hole, *B*, in the damping piston, *D*, to the bottom side of valve disc, *C*, which it raises.

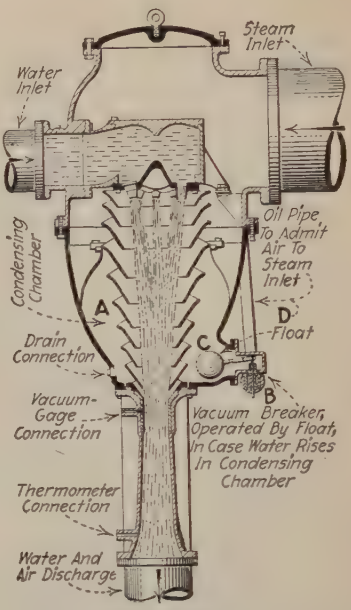


FIG. 186.—Sectional view of the condenser of Fig. 183 showing the vacuum breaker at *B*. If, when the turbine and pump are stopped the water should rise into the condensing chamber, *A*, then float *C* will be thereby raised. This will open the valve *B* which will permit air to flow through *D* into *A*.

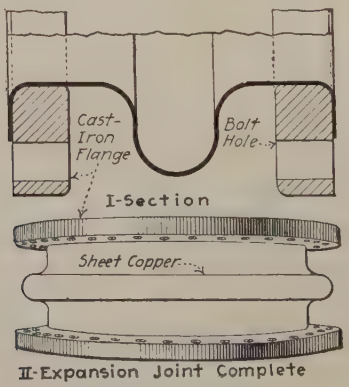


FIG. 187.—Copper expansion joint for low-pressure service.

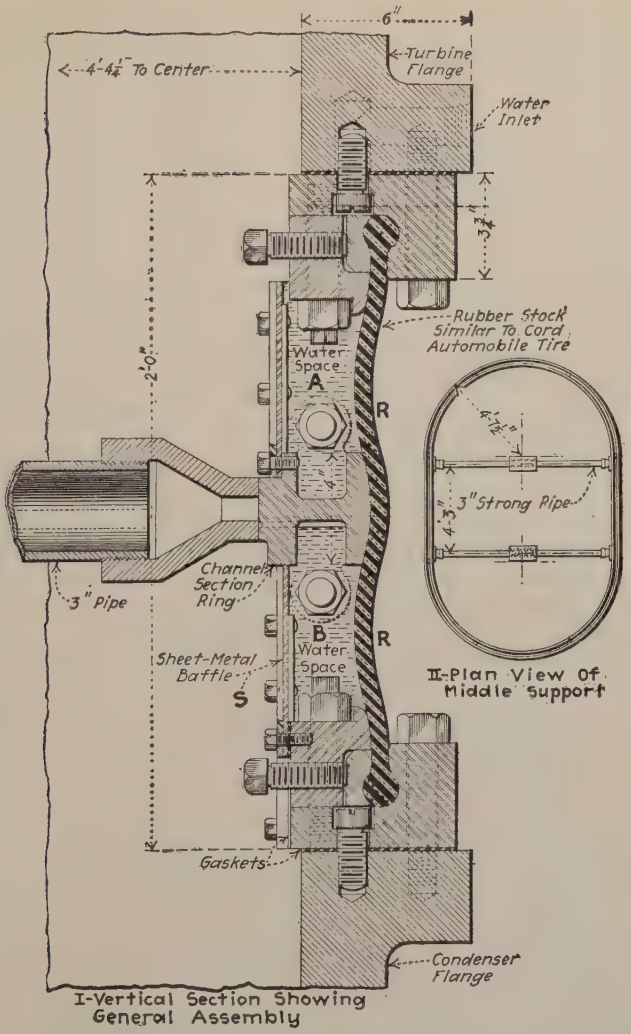


FIG. 188.—Westinghouse rubber expansion joint. The sheet-metal baffle, *S*, provides a smooth passageway for the steam. The rubber member, *R*, is provided with the middle support shown in *II*. Thus, the stresses in *R*, due to the pressure of the atmosphere on the outside of the joint, are small. Member *R*, can be replaced without disturbing any piping or equipment. The spaces, *A* and *B*, between *R* and *S* are so arranged that they may be kept full of water and so protect the rubber against the high-temperature steam whenever the turbine is exhausting against atmospheric pressure—as when starting. Connections are also provided for admitting make-up water to these chambers.

tions should be piped with an atmospheric relief—for free exhaust—valve, (Figs. 182 and 185) in the exhaust line. This value is provided so that, should the condenser fail to function, the turbine may exhaust to the atmosphere. All low-level jet condensers should be fitted with a vacuum breaker (*B*, Fig. 186) to prevent the possibility of water being sucked up into the turbine at any time.

Note.—The Methods Of Connecting Condensers To Turbines are two: (1) *With expansion joints;* it is customary for units smaller than 10,000 kw., and sometimes for larger units, to take care of the upward and downward movement of the condenser by using a flexible expansion joint between it and the turbine. Copper joints (Fig. 187) have been widely used in the past but, due to their short life, they are being replaced by telescoping steel or by rubber joints; see the author's Machinery Foundations and Erection. Fig. 188 shows a rubber expansion joint. (2) *Direct connections* between turbine flange and condenser flange, or with a short nozzle between, are often used on the larger units; the condenser is then mounted on springs so designed that the maximum limits of operating conditions—that is, high vacuum and non-condensing—will not cause a strain on the turbine casing flange which is in excess of the value specified by the turbine manufacturer. Condenser supports are described in the author's Machinery Foundations and Erection.

QUESTIONS ON DIVISION 8

1. What is the function of a regenerator as used with steam turbines? What plant conditions usually call for a regenerator?

2. Draw a sketch to show the construction and operation of a Rateau regenerator. Explain its operation.

3. List the piping accessories with which a regenerator should be equipped and give the reason for each.

4. State briefly under what conditions a regenerator is practical.

5. Describe how boiler-pressure variations may be utilized with a regenerator. Draw sketches to show two methods of utilizing the boiler blowdown in a regenerator.

6. What operating pressure is usually employed in a regenerator? How much above and below this pressure should the pressure be permitted to vary? What is the objection to employing a slight vacuum in the regenerator?

7. Describe the process of equalizing the steam requirements of non-condensing engine and low-pressure turbine.

8. How may the necessary weight of water in a regenerator be computed? State the formulas.

9. What is the purpose of employing a condenser in connection with a turbine? How does the condenser accomplish this purpose?

10. What type of condenser is most generally employed with steam turbines? Why?

11. Explain the process of determining the cooling surface and circulating water requirements for a condenser. What values are considered satisfactory for the temperature difference between the exhaust steam and cold water? Exhaust steam and hot water? For the rate of heat transfer?

12. What are the customary methods of connecting turbines to their condensers?

PROBLEMS ON DIVISION 8

1. What weight of water should be stored in a regenerator which is to serve a 1,500-hp. low-pressure turbine which uses 25 lb. of steam per hp.-hr. if the regenerator pressure may vary between 22 and 25 lb. per sq. in. abs.? The steam supply may be cut off from the regenerator for 3 min. or there may be a momentary supply of 2,000 lb. of steam.

2. If the turbine of Prob. 1 is situated where a liberal supply of cold water is available at an average temperature of 60° F., will it be feasible to operate it at an absolute exhaust pressure of 1.5 in. of mercury column and, if so, what condenser surface and how much circulating water will be required?

DIVISION 9

HIGH-PRESSURE, BLEEDER, MIXED-PRESSURE AND EXHAUST-STEAM TURBINES

175. The Extensive Use Of The Steam Turbine In Modern Industry Is Due Partly To Its Adaptibility To All Steam Conditions. (See Table 29 for classification of turbines according to steam conditions.) The relations of the different kinds of turbines to the power-plant steam pressures is shown graphically in Fig. 189. Steam turbines are used not only for the

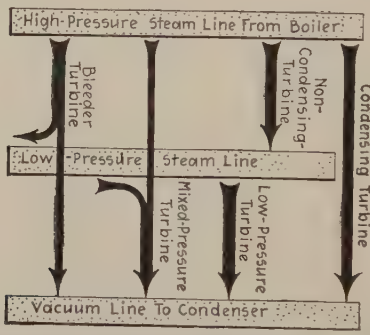


FIG. 189.—Showing the relation of turbines designed for various steam conditions to the various steam pressures which are used in power plants.

high-pressure condensing and non-condensing services for which steam engines are applicable, but they are also applied to a number of "special" services—such as bleeder, mixed-pressure and low-pressure services—as will be later explained (see definitions of the "special" turbines in Secs. 35 to 37). The value of the turbine for these special services is due largely to the fact that it derives considerably more power from low-pressure steam in condensing service than does a steam engine.

NOTE.—WHEN BOTH HEAT AND POWER ARE SUPPLIED BY THE SAME POWER PLANT, it is economical to generate the steam (which will be

required for heating) at high pressure and run it through a relatively inefficient non-condensing engine or turbine before it is delivered to the heating system. When this is done the power thus secured from the non-condensing engine or turbine is a sort of byproduct; and only a small amount of fuel is burned, in addition to that which would be required for heating alone, for its production. On the other hand, if more power is required than can be thus obtained, this additional power can, in most instances, be most economically obtained with a condensing steam turbine. The "special" turbine is particularly useful in improving the combined economy of a heating and power plant.

176. Table Showing How The Requirements Of Any Given Set Of Steam Conditions May Be Fulfilled By A Turbine Unit.

Case	Exhaust steam available	Exhaust steam needed	Turbine used
1	None	None	Condensing turbine
2	Always more than turbine needs	None	Low-pressure turbine
3	Sometimes more—sometimes less than turbine needs	None	Mixed-pressure turbine or low-pressure turbine and regenerator
4	None	Always more than turbine will supply	Non-condensing turbine and reducing valve
5	None	Sometimes more—sometimes less than turbine will supply	Bleeder turbine and reducing valve

Note.—The Requirements In The Above Table Are Assumed To Be Normal—Not Emergency Requirements. In case 3, if the turbine load is only occasionally too great for the steam supply, it may be advisable to "bleed" the live steam line (use a reducing valve) and use a low-presure turbine rather than to install a mixed-pressure turbine. In case 5, if the surplus of exhaust is only occasional, it may be more economical to use a non-condensing turbine than to employ a condenser and a bleeder turbine. Where it is indicated above that no exhaust steam is needed, it is meant that there are auxiliaries or other

equipment in the plant which will supply what *is* needed—none is needed from the turbine under consideration.

Note.—A Condenser Is Always Necessary For "Condensing," "Low-pressure," and "Bleeder Turbines." see Fig. 189. The operation of turbines of these types without a condenser would be an impossibility. Condensers are generally used also with mixed-pressure turbines.

177. The Relative Amounts Of Heat Energy Which Are Theoretically Available For Turbines Operating Under

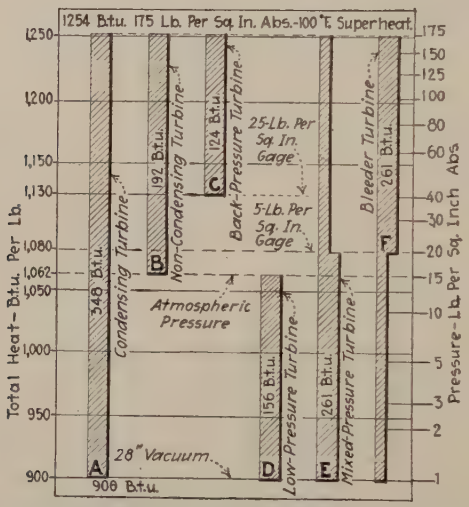


Fig. 190.—Amounts of heat available (given up by adiabatic expansion) from each pound of steam for conversion into work by turbines of various types operating under typical steam conditions. (It is assumed that, in the mixed-pressure turbine, *E*, ½ lb. of steam is admitted at boiler pressure; the other ½ lb. is run through a non-condensing engine and admitted to the turbine at 20 lb. per sq. in. absolute. In the bleeder turbine, *F*, ½ lb. of steam is bled at 20 lb. per sq. in. absolute.)

Different Steam Conditions may be understood by a study of Fig. 190; see also Fig. 261. These values of the heat available hold only for the steam conditions indicated, but these conditions are typical. The amount of heat which is actually converted into work is about one-half to three-fourths (depending on the size of the unit; see Fig. 20) of the values given in Fig. 190. It is assumed in Fig. 190 that the low-pressure turbine operates on the exhaust from the high-pressure turbine. Therefore the low-pressure turbine does not receive

a full pound of dry steam for each pound of steam admitted to the high-pressure turbine. If dry steam is used by the low-pressure turbine, the available heat at this vacuum is 171 B.t.u. per lb. but, if the moisture were removed from the steam by using a separator, practically nothing would be gained or lost. In actual practice a large condensing turbine develops about twice as much power from the same steam as does a non-condensing turbine, or as much as does a combination of a high- and low-pressure turbine together, in which the high-pressure turbine exhausts into the low-pressure turbine. For methods of calculating the available energy, efficiency, etc. under different steam conditions, see Secs. 10 and 13.

178. A High-pressure Non-condensing Turbine Is Especially Useful under the following conditions, see Sec. 34 for definition: (1) *When used in conjunction with a low-pressure or exhaust-steam turbine as part of a compound unit.* (2) *When there is usually a demand for all the exhaust steam which is produced by the turbine in driving its load.* (3) *When lack of space, water, or other considerations render condensing operation infeasible.* Non-condensing turbines find extensive application for auxiliary drives (*A* and *B* Fig. 206) and small power purposes where the steam consumption is of minor importance or where the exhaust may be used for feed-water heating. The non-condensing turbine is seldom, except in small capacities, used alone as a prime mover because it develops only about one-half of the power which a condensing turbine will develop on the same amount of steam.

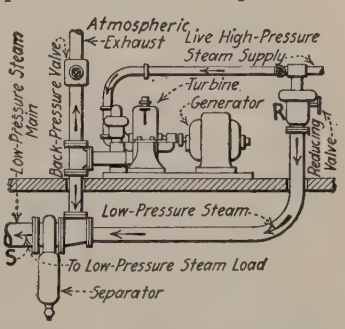


FIG. 191.—A high-pressure non-condensing turbine, *T*, piped for service where there is demand for more low-pressure steam, *S*, than is supplied by the turbine.

179. Turbines Of The Simpler Types Are Usually Used For Non-condensing Service Where All Of The Exhaust Steam Is Useful For Heating (Fig. 191). Under these conditions, the steam consumption is of comparatively little importance. Velocity-and-pressure-staged turbines (Sec. 61) having one or two pressure stages are widely

used for this sort of service. Bucket-wheel and impulse re-entry turbines of the axial-flow type are also widely used. Turbines of these types are relatively inexpensive in proportion to the power which they develop but have relatively high water rates; see Div. 14.

180. High-pressure Condensing Turbines Are Useful Whenever A Single Unit Is Desired Solely For The Development Of Power.—They (Fig. 192) are built in sizes up to about

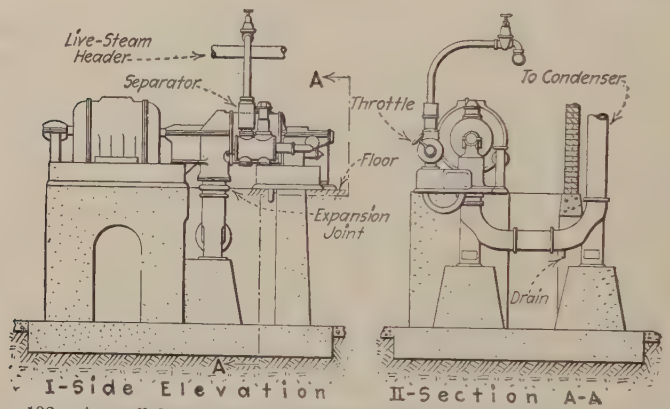


FIG. 192.—A small high-pressure condensing turbine piped for service. Usually, the most desirable location for the condenser is directly under the turbine rather than at some distant location, which is indicated by the above illustration.

35,000 kw. as single-cylinder units and up to 70,000 kw. as compound units. The condensing turbine has become the accepted prime mover for all large modern electric generating and low-head pumping stations where steam power is used. The reason for this is the high efficiency and large power output of these turbines in proportion to their size and cost. See Div. 14 for economies.

NOTE.—THE CONSTRUCTION OF CONDENSING TURBINES varies greatly with the conditions. Single-stage impulse turbines of the single entry and re-entry types are sometimes operated condensing. Large condensing turbines for central stations are multi-stage turbines of impulse, reaction, or impulse-and-reaction types.

181. A Bleeder Turbine (Sec. 37 and Figs. 193 and 194) may be considered as a high-pressure turbine which can

operate: (1) *Condensing*, (2) *non-condensing*, or (3) *partly condensing and partly non-condensing at the same time.* Under

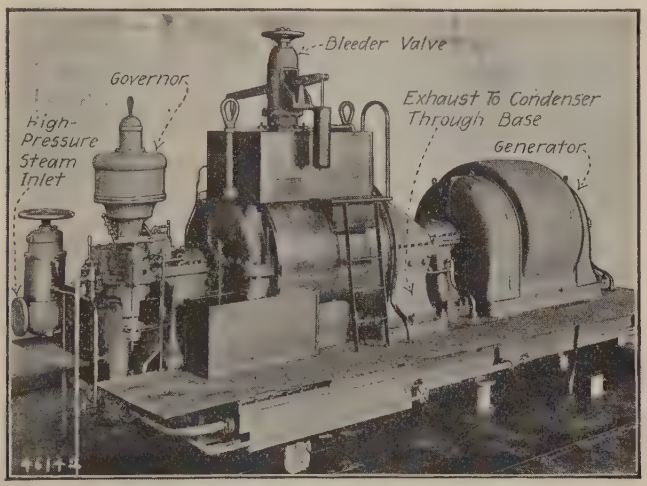


FIG. 193.—Westinghouse automatic bleeder turbine—single-flow type. A vertical section of a similar turbine is shown in Fig. 194.

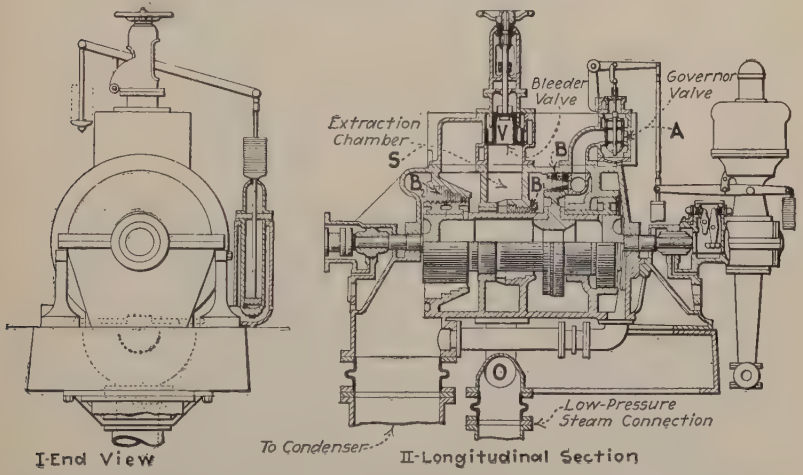


FIG. 194.—Vertical section and end elevation of a 1,500-kw. *Westinghouse* bleeder turbine.

some conditions it will operate almost wholly as a condensing unit; under others, almost wholly as a non-condensing unit.

The turbine is so automatically controlled that it will: (1) Utilize and exhaust into the heating system all steam which is admitted by its governor and which is required in the heating system; if enough steam for the heating-system requirements is not admitted by the turbine governor, high-pressure steam may be automatically bypassed into the heating system through a reducing valve, *R* Fig. 191. (2) Utilize and condense all of the steam which is admitted by its governor in excess of that required by the heating system.

Explanation.—Consider that the bleeder turbine (*T*, Fig. 195) is installed in a plant which requires power all the year for lights and small

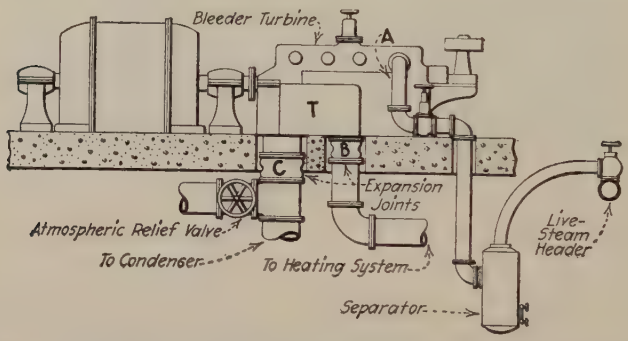


Fig. 195.—A bleeder turbine installed to supply a low-pressure main with steam and condense the exhaust which is not needed for heating.

motors and requires an amount of heat which varies greatly with the changes in the weather. The bleeder turbine is supplied with live steam at *A*. Low-pressure steam for heating is withdrawn at *B*. The steam, in passing from *A* to *B* in the turbine, does work which is useful in generating power. The steam which is not needed for heating passes on through *C* to the condenser, thus doing more work. In this way the heating and power requirements of the plant are satisfied and all of the steam is used as economically as is reasonably possible.

182. The Governing Of A Bleeder Turbine And The Proper Distribution Of Steam In It require a regular speed governor and a bleeder valve. The turbine and governor (see Div. 6) are very similar to an ordinary condensing turbine and governor. A bleeder valve (*V*, Fig. 194 and Fig. 196)

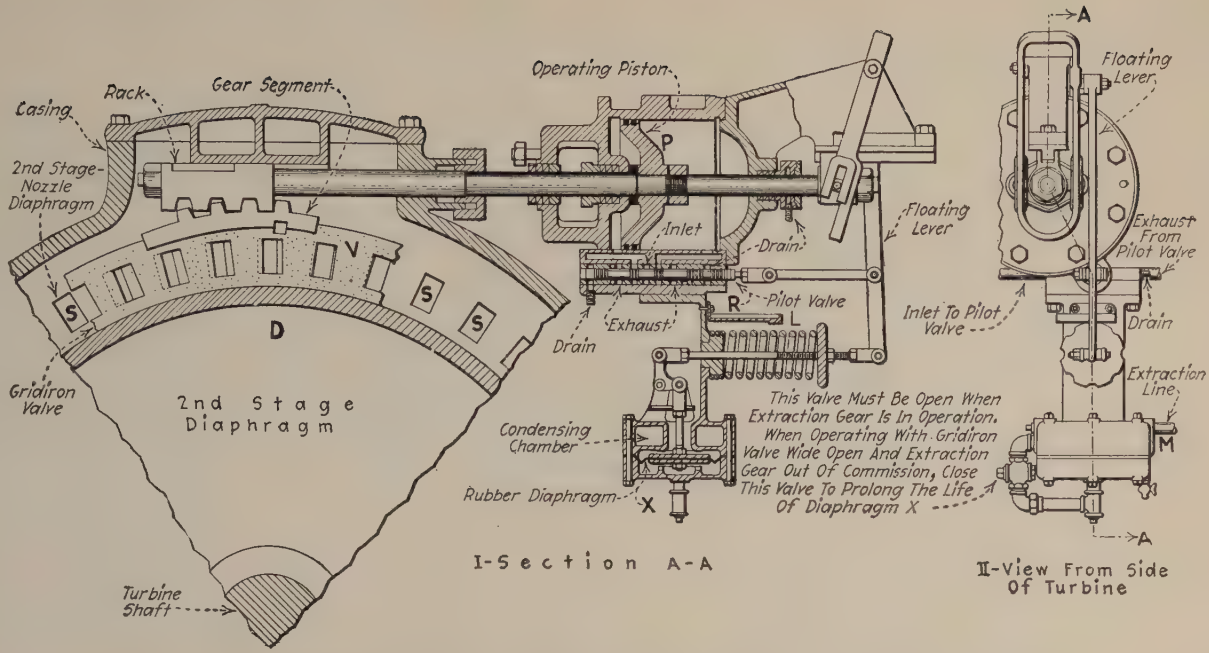


FIG. 196.—Diagram of bleeder or extraction valve of a *General Electric Co.* Curtis bleeder turbine.

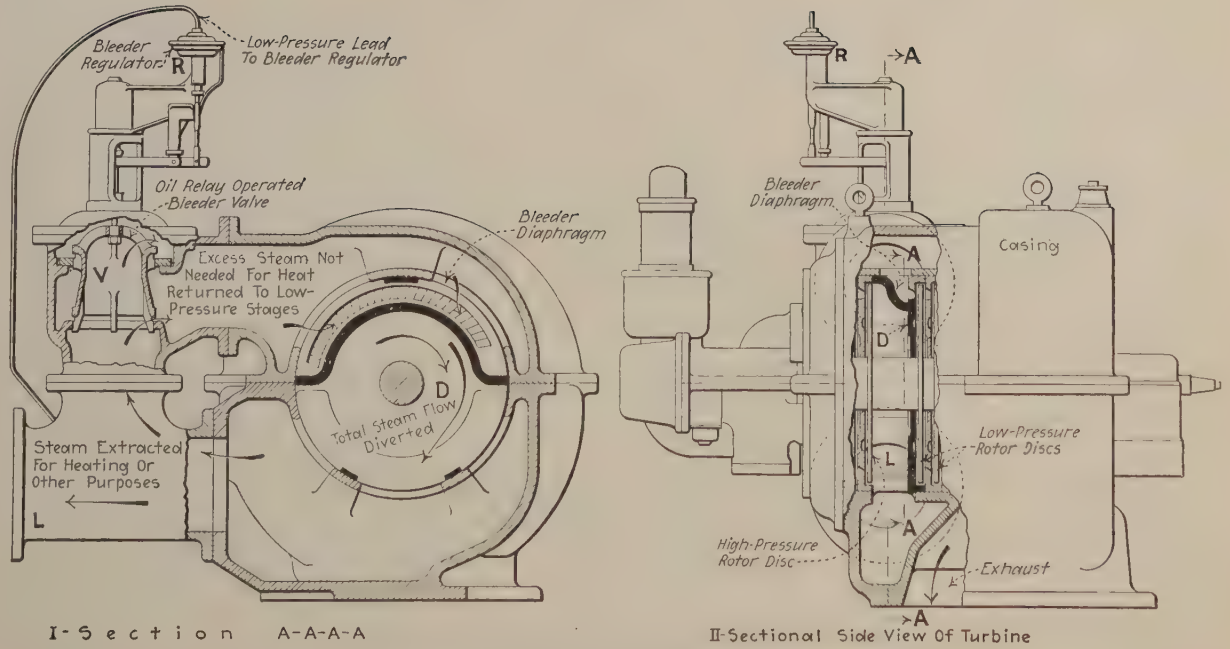


FIG. 196A.—Diagrams showing bleeder mechanism of the *Terry* turbine.

must however be provided in order to bleed or extract sufficient steam to maintain a desired pressure in the heating system.

Explanation.—In the *Westinghouse* turbine shown in Fig. 194, the steam is admitted through governor valve, A, and flows through impulse blading, B, and high-pressure reaction blading, B_1. Then, if the pressure is low in the low-pressure line connected at O, low-pressure steam is

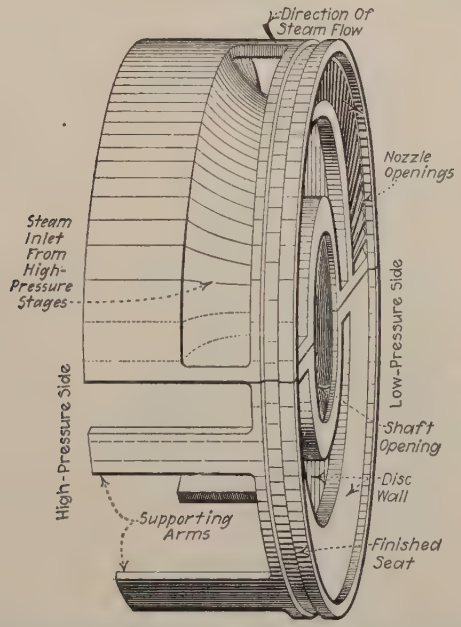


Fig. 196*B*.—Bleeder diaphragm of the *Terry* turbine which completely stops the steam flow through the turbine, diverting it to the bleeder line. Steam returned through the bleeder valve (Fig. 196*A*) enters the nozzles in the upper half of this diaphragm and then passes on through the turbine.

withdrawn through that passage. If the steam pressure increases in O, the valve, V, which is similar in its action to a weight-loaded safety valve, opens and allows low-pressure steam to flow through the low-pressure blading, B_2, to the condenser. A check or non-return valve is always provided in the low-pressure steam line to prevent flow of steam back to the turbine.

Note.—The General Electric Co. Bleeder Mechanism is shown in Fig. 196. The bleeder or extraction valve consists of a diaphragm, D, placed across the turbine cylinder at the point where it is desired to bleed the turbine, and a valve disc, V. The diaphragm and disc are so

arranged that, as the disc is rotated part of a revolution, the slots, *S*, in the diaphragm (through which steam is admitted to various nozzle sections) are uncovered successively. That is, a slight rotation will uncover one slot; a larger rotation will uncover two, three, or all of the slots. The rotation of the disc is controlled by the piston and relay mechanism, *PRX*. Steam from the low-pressure line is admitted behind the spring-opposed diaphragm, *X*. The movement of this diaphragm operates the piston, *P*, through the oil-relay valve, *R*. In this way the opening of the relay valve is controlled by the steam pressure in the low-pressure line. By adjusting spring, *L*, this pressure may be maintained at any reasonable desired value. The advantage claimed for this method of extraction over that of Fig. 194 is that with the Fig. 196

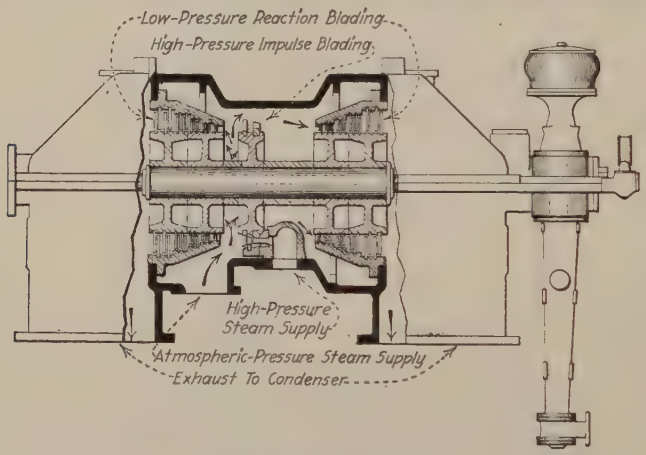


Fig. 197.—Vertical section of a mixed-pressure turbine.

method there is little throttling action in the bleeder valve since most of the slots, *S*, are always either wide open or tightly closed.

Note.—The Bleeder Mechanism Of The Terry Turbine is shown in Fig. 196*A*. It differs from the bleeder mechanisms just described in that the steam flow through the turbine is completely stopped off by a special diaphragm, Fig. 196*B*. The steam, after flowing through the first stages of the turbine, is diverted by this diaphragm (*D*, Fig. 196*A*) into the low-pressure steam pipe, *L*. Should the pressure in this pipe become too great, it will displace a diaphragm in the regulator, *R*, and thereby open an oil-relay valve. Oil will then flow through the relay valve to a piston on the same rod as the bleeder valve, *V*, thus opening *V*. Steam will then flow through *V*, again into the turbine—now through the low pressure stages. Should the pressure in *L* become too low, the reverse action takes place—valve *V* is closed. The valve, *V*, is so proportioned that, should its operating mechanism become in-

active, it will automatically open at a predetermined pressure in L, thus avoiding dangers due to excess pressure.

183. A Mixed-pressure Turbine (Sec. 36 and Fig. 197) may be considered as a combination, in a single machine, of a high-pressure and a low-pressure condensing turbine. A mixed-pressure turbine is so controlled that no high-pressure steam will be used unless the low-pressure steam supply is inadequate for the power requirements of the turbine at that instant.

Explanation.—Consider that the mixed-pressure turbine (T, Fig. 198) is installed to utilize the exhaust steam from the engine, E. Exhaust

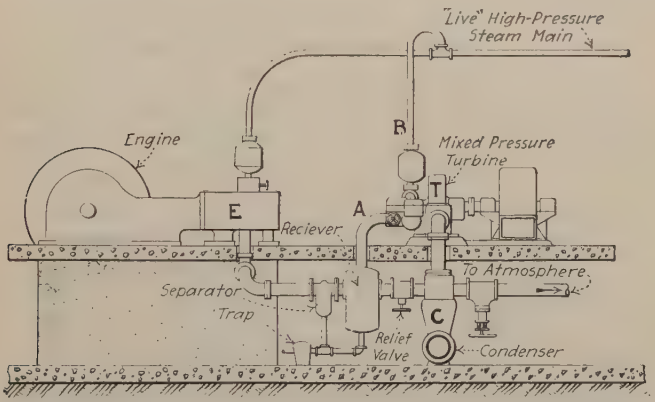


Fig. 198.—Mixed-pressure turbine installed for service in connection with a reciprocating engine.

steam is admitted to the turbine at A and flows through it to condenser, C. If the load on the engine is heavy and that on the turbine is light, the turbine runs as a low-pressure turbine, and the surplus exhaust steam from the engine is condensed. Now suppose that the load on the engine becomes very light and that on the turbine becomes very heavy. The turbine will then derive little power from the engine exhaust and would stop if no other source of power were available. But the governor of the turbine then admits high-pressure steam at B which flows through all of the stages of the turbine. The turbine will then derive most of its power from the high-pressure "live" steam just as does a high-pressure condensing turbine.

184. The Functions Of A Governor For A Mixed-pressure Turbine (see Div. 6) are: (1) *To admit all available low-pressure*

steam provided it is all required by the turbine in delivering its load. (2) *To shut off the low-pressure steam if more than sufficient for the load.* (3) *To admit just sufficient additional high-pressure steam to carry the load when the low-pressure steam supply is inadequate.*

Explanation.—These functions may all be accomplished very simply by the arrangement shown diagrammatically in Fig. 199. The governor is shown in *I* (Fig. 199) in the no-load position, with the weights or balls raised. When load is applied to the turbine and its speed decreases, the

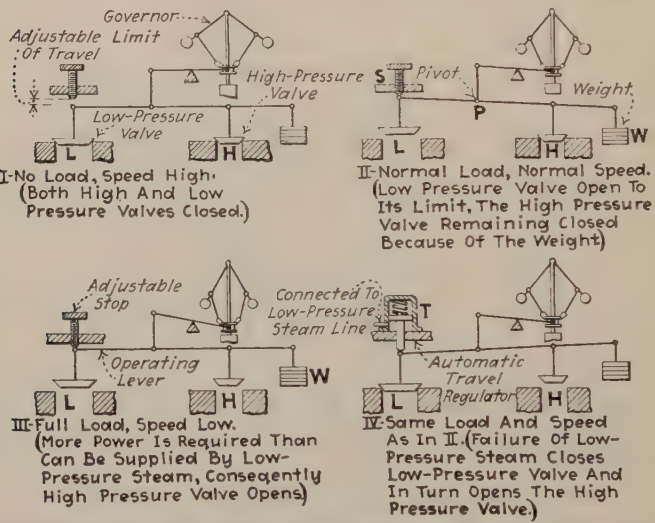

Governor
Adjustable Limit Of Travel
High-Pressure Valve
Low-Pressure Valve
L
H
I-No Load, Speed High. (Both High And Low Pressure Valves Closed.)
S
Pivot
Weight
P
W
II-Normal Load, Normal Speed. (Low Pressure Valve Open To Its Limit, The High Pressure Valve Remaining Closed Because Of The Weight)
Adjustable Stop
Operating Lever
III-Full Load, Speed Low. (More Power Is Required Than Can Be Supplied By Low-Pressure Steam, Consequently High Pressure Valve Opens)
Connected To Low-Pressure Steam Line
T
Automatic Travel Regulator
IV-Same Load And Speed As In II. (Failure Of Low-Pressure Steam Closes Low-Pressure Valve And In Turn Opens The High Pressure Valve.)

Fig. 199.—Diagrammatic representation of the operation of a governor for a mixed-pressure turbine. (*Terry Turbine Co.*)

balls drop, as in *II*, and lift pivot, *P*. Due to the weight, *W*, the movement of *P* lifts the low-pressure valve, *L*, but high-pressure valve, *H*, is held on its seat as in *II* (Fig. 199). After the low-pressure valve has traveled as much as the adjustable stop, *S*, will permit, as in *III*, further movement of the governor lifts high-pressure valve, *H*, against the downward force of *W*. If it is desired to maintain a certain back pressure in the low-pressure steam line, an automatic travel regulator, *T* (Fig. 199, *IV*) must be employed. This consists of a cylinder containing a spring-loaded piston. If there is no pressure in the lower part of *T*, no travel of the low-pressure valve is permitted and the turbine runs on high-pressure steam. But if a pressure is produced in the lower part of *T*, the lifting of *L* is permitted so that low-pressure steam is admitted to the

turbine. An actual mixed-pressure governor valve is shown in Fig. 200. This arrangement never closes the low-pressure valve when there is load on the turbine. If it is desired to maintain a back pressure, a constant-pressure valve (Fig. 201) must be used. This valve also acts as a check to prevent a flow of steam from the turbine to the low-pressure line in

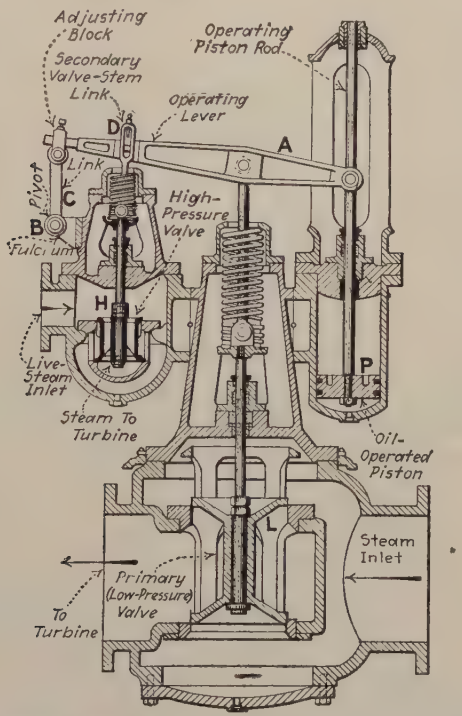


Fig. 200.—Governing valves of a mixed-pressure turbine. As oil is admitted from the relay valve (not shown) to the under side of the piston, P, lever A is rotated upward and to the left with the link, C, which is pivoted at B. This raises and opens the low-pressure valve, L. At a certain point in the upward motion of P, the lost motion in link D is taken up. Further upward motion of P will also open the high-pressure valve, H. No provision is made in this governing mechanism for keeping L closed when the pressure in the low-pressure steam supply-pipe becomes abnormally low.

case the low-pressure steam supply fails. See also Fig. 202.

Note.—Low-pressure Steam Is Sometimes Supplied To The Later Stages Of An Ordinary Condensing Turbine Through Only A Flow Valve (Fig. 201).—Turbines which are so arranged are not generally called mixed-pressure turbines although they really function as such. The low-pressure steam is admitted by the flow valve whenever the pressure in the supply pipe (the exhaust pipe of the non-condensing

equipment) exceeds a predetermined value. There is no speed-governor valve to control the admission of the low-pressure steam. Hence, should such a turbine be run under very light load at a time when the low-pressure supply is plentiful, the turbine may run at a speed well

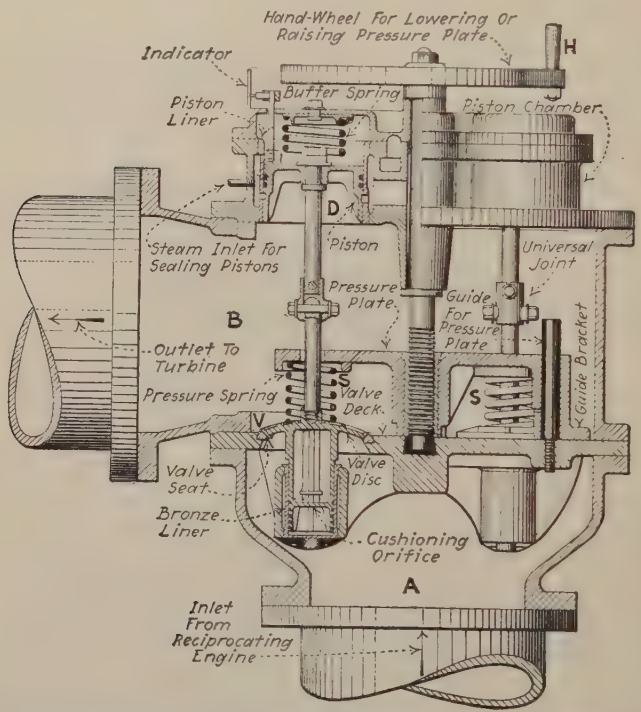


FIG. 201.—Cochrane "constant-pressure" multiport flow valve (reducing valve) used at the low-pressure inlet of a low- or mixed-pressure turbine. This valve is used to maintain a constant back pressure on a non-condensing unit. This valve may be set to maintain the desired constant pressure by turning *H*, which changes the compression of the springs, *S*. If the pressure in *A* falls below that for which the valve is set, steam pressure in *B* lowers valve discs, *V*, and shuts off steam from *B*. If the pressure in *A* increases above the pressure for which the valve is set, the pressure in *A* lifts the valves, *V*, against the springs, *S*. At *D* is a dashpot which prevents chattering and above it is a buffer spring.

above normal. To prevent such overspeed damage to the turbine, the low-pressure supply is shut off by the automatic overspeed governor when the turbine's speed reaches the value at which this emergency governor is set to operate. Hence, *such turbines should be used only where there is very little likelyhood that the low-pressure steam supply will ever exceed the requirements of the minimum load on the turbine.*

185. Mixed-pressure Turbines Are Sometimes Used For Auxiliary Drives. Figure 203 shows mixed-pressure main turbine, T, and auxiliary turbine, A, so connected that they may derive steam from the receiver, R, of a compound engine. These turbines running condensing are considerably more

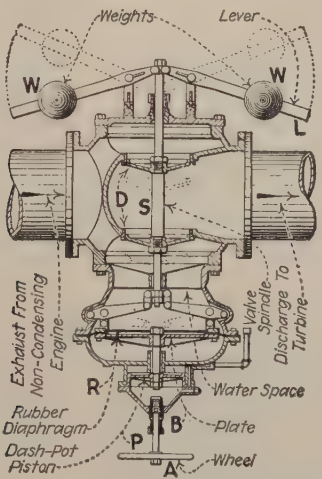


FIG. 202.—*Schutte & Koerting* automatic flow regulating valve. This valve is, in function, similar to that of Fig. 201. This valve will, however, maintain a constant pressure on its supply side regardless of the pressure on its discharge side and without manual adjustment. On the other hand, this valve does not serve as a check valve whereas that of Fig. 201 does. The rubber diaphragm, R, is supported by plate, B, and is submerged in water to protect it from the hot steam. Multiplying levers connect B with the valve spindle, S. The valve is shown in the closed position, which it normally occupies when the pressure above R is less than about 16 lb. per sq. in. abs. A greater pressure above R will cause it to lower the valve discs, D, and raise the weights, W, on levers, L. Steam may then pass through the valve to the turbine. Should the pressure above R fall below 16 lb. per sq. in. gage the valve will be closed by the weights, W. The valve may be blocked in the closed position by screwing up wheel A.

economical than the low-pressure cylinder, L, of the engine. Thus for most loads on the engine, auxiliary power is secured with a negligible amount of extra steam. When there is an overload on the engine or when the engine is not running, live steam may be admitted, through M and N, to the main and the auxiliary turbines.

186. There Are A Number Of Automatic Or Partly-automatic Methods Of Balancing The Heat And Power Requirements Of A Steam-turbine Power Plant.—In some of these, (Sec. 184)

the automatic balancing is accomplished by employing valves which are sensitive to variations in the pressure of the low-pressure steam. In others, the balancing is accomplished by electrical or mechanical means. See examples below.

Note.—"Heat Balance" Or A Balance Between The Power Required For Auxiliary Drives And Heat Required For Feed-water And Other Heating is an important consideration in most

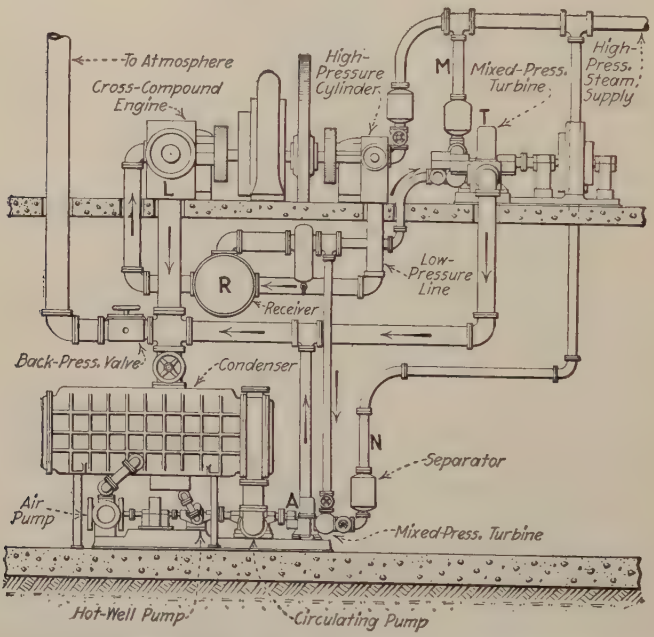


Fig. 203.—Mixed-pressure turbines, *T* and *A*, arranged to operate on steam from the receiver, *R*, of a cross-compound condensing engine.

plants. From about 2 to 10 per cent. of the steam generated by the boiler is generally required, directly or indirectly, to drive the auxiliaries of the power plant. About 5 to 8 per cent of the steam generated by the boiler may—after it has been used in some non-condensing engine or turbine—be profitably used for heating the feed water. Sometimes, therefore, if all the auxiliaries are steam driven, they will supply enough —or more than enough—exhaust steam for feed-water heating. For maximum economy, there should, theoretically, be just enough exhaust steam available for feed-water heating but there should be no waste of exhaust. The temperature to which the feed water should be heated

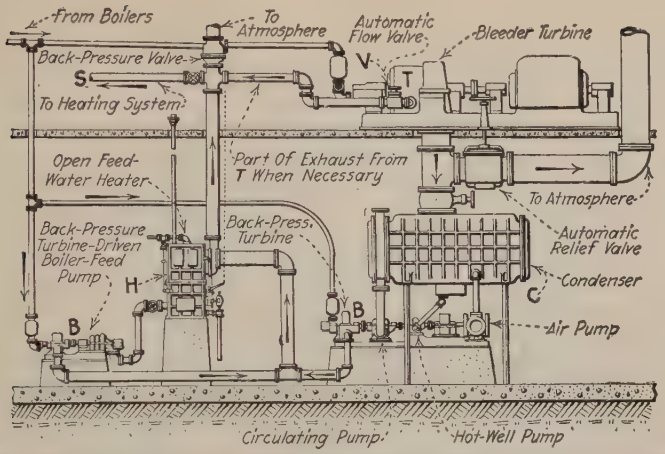


FIG. 204.—Heat-balance system with bleeder turbine prime mover and back-pressure-turbine driven auxiliaries. (*De Laval Steam Turbine Co.*) The back-pressure turbines, *B* and *B*, operate on live steam from the boilers. They exhaust into the feed-water heater, *H*, against a back-pressure. The flow valve, *V*, permits steam to flow from the extraction chamber of main turbine, *T*, into the heating system whenever the difference between the pressures in the two exceeds the value for which *V* is set. Thus as the load on *T* varies, the pressure in the heating system may also vary unless *V* is adjusted by the operator. For periods when the load on *T* is very small, a reducing valve (not shown) may be necessary to admit live steam to *S*.

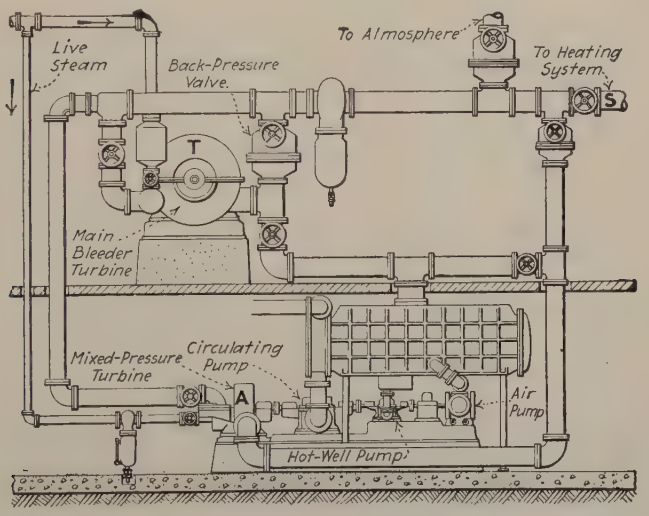


FIG. 205.—Heat balance system in which two bleeder prime-mover turbines, *T*, (only one is shown) are used and in which the auxiliary drive turbines, *A*, are of the mixed-pressure type.

with exhaust steam is about 210° F. where no economizer is used. Where an economizer *is* used, the feed-water temperature should vary from about 210 to 150° F. as the water flows from the exhaust-steam heater. See the author's Steam Power Plant Auxiliaries and Accessories.

Example 1.—In the arrangement of Fig. 204, both the bleeder turbine, *T*, and the back-pressure-turbine (Sec. 34) driven auxiliaries, *B* and *B*, are connected to supply steam to the feed-water heater, *H*, and to the steam-heating system, *S*. When the steam from the auxiliaries is ample for all steam-heating requirements, no steam will flow from the bleeder turbine, *T*, to the heating system. All of the steam which the turbine, *T*, then uses will be condensed in *C*. Thus the bleeder turbine itself furnishes an automatic means of keeping the heating requirements and the low-pressure steam supply balanced.

Example 2.—Figure 205 shows an arrangement in which the main turbine, *T*, is a bleeder and the auxiliary drives, *A*, are mixed-pressure turbines. For very heavy heating loads, the main turbine is run entirely non-condensing and exhausts to the heating system, *S*. When there is no heating load, the main and auxiliary turbines are both run entirely condensing. When there is a moderate heating load, steam is bled from the main turbine to supply both the heating system and the mixed-pressure auxiliary turbine with low-pressure steam. If the power load is increased so that it cannot all be handled thus, the auxiliary turbine may run entirely on high-pressure steam and exhaust to the heating system. This will permit condensing all of the exhaust from the main turbine so that its maximum power will be developed.

187. An Electrical Method Of Effecting An Exhaust-steam Heat Balance In A Power Plant is shown in Fig. 206 (from *Power*, Sept. 6, 1921). This method is applicable either for plants which are used for developing electrical energy only or for combined heating and power plants.

Explanation.—The main turbine, *T*, is operated condensing. In order that the power-plant lighting and motor drives may not be affected by trouble in the main electric system, a non-condensing house turbo-alternator, *H*, is employed to generate the necessary electrical energy which is used in the power plant itself. The motors, *M*, form part of the electrical load on *H*. The exhaust steam from *H* is piped to the barometric feed-water heater *F*. But, since the electrical load on *H* cannot readily be varied, it is obvious that the amount of exhaust steam for feed-water heating will be nearly constant unless some variable load is connected to *H*. If *H* were paralleled with the main generator (by connecting it to the main bus bars), then the load on *H* could be varied by varying its governor-spring tension (Div. 6) thus causing it to furnish more or less power to the main bus bars But this would place the power-plant lighting and motor drives subject to shut down due to trouble on

the main lines. To obviate this possibility, the motor-generator G is introduced as a connecting link between H and the main lines.

The temperature in the feed tank W is recorded by a remote-reading thermometer on the switchboard. The switchboard operator, by manipulating the synchronizing motor on H may then cause H to

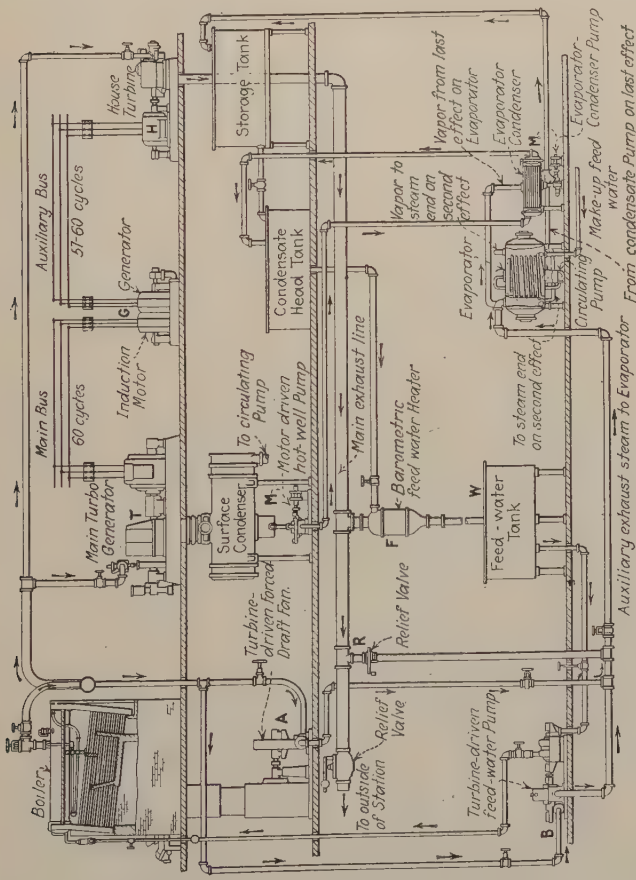


FIG. 206.—Diagram of power-plant equipment in which several small high-pressure non-condensing turbines are piped for service. (This plant is designed to maintain an exhaust-steam heat balance.)

deliver power to or the house system to take power from the main bus (through G) and thus exhaust more or less steam as required for feed-water heating. A definite feed-water temperature, which has been found most economical, may thus be maintained. The exhaust steam from the non-condensing turbines, A and B, which drive the auxiliaries, is used for distilling make-up water. Any exhaust which is not thus used flows through the relief valve, R, to the heater, F.

188. The First Costs Of Mixed-pressure And Bleeder Turbines Are Relatively Low compared to those of separate equipment for the functions which these turbines perform. A bleeder turbine takes the place of a condensing and a non-condensing turbine. It also furnishes automatic means of conserving steam. A mixed-pressure turbine may take the place of an exhaust-steam turbine and a high-pressure turbine. Considered in another way, the mixed-pressure turbine develops power from exhaust steam and obviates the necessity for a regenerator by drawing live steam when the supply of exhaust is low. The cost of this live steam may often be neglected because the times when it is used are those intervals just after the non-condensing equipment has been shut down—at such times there is likely to be a surplus of steam and the safety valves of the boilers would blow if no steam were drawn from the boilers.

NOTE.—THE SPEED REGULATION OF MIXED-PRESSURE TURBINES AND BLEEDER TURBINES (see note under Sec. 125 for definition of "speed regulation") is ordinarily much greater than that of other turbines. In bleeder turbines, the governor valve must open somewhat wider than in ordinary turbines to admit sufficient steam to develop the full power of the unit when the bleeding is heavy. This necessitates more travel of the governor and valve and more variation in speed. In mixed-pressure turbines, the governor gear must travel far enough to open the low-pressure valve and far enough in addition to open the high-pressure valve when there is little exhaust steam. This travel requires a greater governor movement than would be required to admit steam from a single source. Also the speed regulation of mixed-pressure and of bleeder turbines is likely to be slightly different when considerable low-pressure steam is being used or extracted from that when little low-pressure steam is being used or extracted.

189. The Economies Of Bleeder And Mixed-pressure Turbines are calculated from two different standpoints: (1) *A technical standpoint.* From a technical standpoint, the economies of mixed pressure and bleeder turbines are most conveniently calculated on a basis of available heat and efficiency ratio as in Sec. 15. The efficiency ratio of these turbines and of low-pressure turbines when operating near the capacity for which they are designed is about the same as that of high-pressure condensing and non-condensing turbines

of the same capacities. (2) *A commercial standpoint.* An example of how the steam consumption of a *bleeder* turbine may be considered commercially is shown in Fig. 207; the turbine is, from this standpoint considered to *consume* only that steam which it condenses. The consumption is considered to be the net consumption, or that fed to the turbine

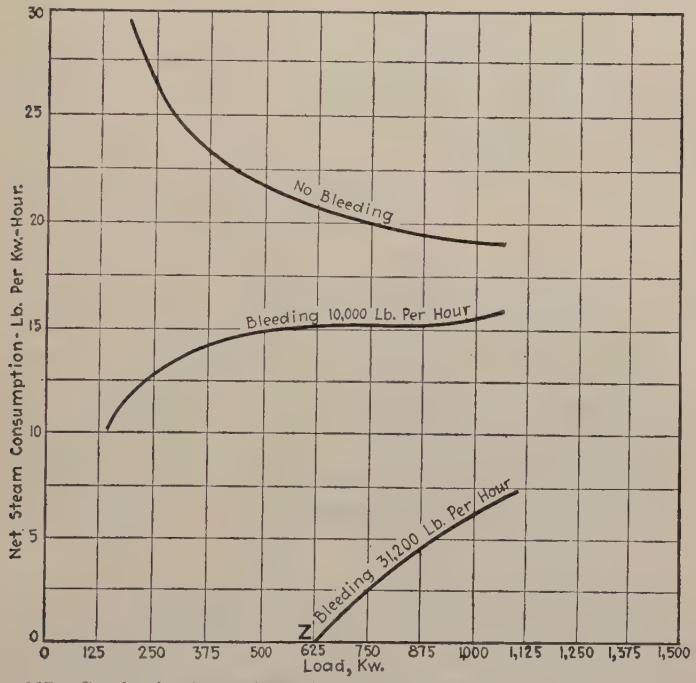


Fig. 207.—Graphs showing variation in commercial economy of a 1,000-kw. bleeder turbine with variations in load and rate of bleeding.

at boiler pressure minus that bled from the turbine at a low back pressure. For commercial purposes this assumption is not much in error because the steam which is bled has 90 to 95 per cent. as much heat as has the high-pressure steam. When the turbine is bled heavily and is carrying a light load, its "commercial" steam consumption may, on this basis as shown, Z (Fig. 207), be practically zero. Similarly a *mixed-pressure* turbine would in some instances, where there is a surplus of exhaust steam from non-condensing equipment,

be charged with only the live steam which it used. Then its steam consumption, from a commercial standpoint, might be zero most of the time.

190. To Compute The Approximate Rate At Which A Mixed-pressure Or Bleeder Turbine Consumes High-pressure Or Live Steam, use the following formula:

(30) $\mathbf{W}_{B1} =$

$$\frac{1}{\mathbf{H}_1 - \mathbf{H}_2}\left[\frac{2{,}545}{\mathbf{E}_r} - \mathbf{W}_{B2}(\mathbf{H}_1' - \mathbf{H}_2')\right] \text{ (lb. per hp. hr.)}$$

Wherein: $\mathbf{W}_{B1}$ = the weight of high-pressure steam, in pounds per brake horsepower-hour, which passes through all of the stages of the turbine. $\mathbf{W}_{B2}$ = the weight of low-pressure steam which is admitted to a mixed-pressure turbine or which is extracted from a bleeder turbine, in pounds per brake horsepower-hour. $\mathbf{H}_1$ and $\mathbf{H}_2$ = respectively, the initial and final total heats per pound of steam before and after adiabatic expansions, of the high-pressure steam, corresponding to the weight $\mathbf{W}_{B1}$. $\mathbf{H}_1'$ and $\mathbf{H}_2'$ = respectively, the initial and final heats per pound of steam, corresponding to $\mathbf{W}_{B2}$. $\mathbf{E}_r$ = the efficiency ratio, or ratio of *the actual efficiency* to *that of the ideal Rankine cycle;* this is the value plotted in Fig. 20. The heat values are found on the graphs of Fig. 15, exactly as explained in Sec. 15 for regular high-pressure turbines.

Example.—A 2,000 hp. mixed-pressure turbine consumes at full load, 9 lb. of steam per horsepower-hour at atmospheric pressure. The condenser maintains 28.5 in. of vacuum. How much high-pressure steam at 175 lb. per sq. in. gage and 100° F. superheat will it also consume at full load? Solution.—From Fig. 20, $\mathbf{E}_r = 0.65$. Also, from Fig. 15, $\mathbf{H}_1 = 1{,}256$; $\mathbf{H}_2 = 888$; $\mathbf{H}_1' = 1{,}150$; $\mathbf{H}_2' = 965$. Hence, by For. (30): $\mathbf{W}_{B1} = [1/(\mathbf{H}_1 - \mathbf{H}_2)][(2{,}545/\mathbf{E}_r) - \mathbf{W}_{B2}(\mathbf{H}_1' - \mathbf{H}_2')] = [1 \div (1{,}256 - 888)] \times \{(2{,}545 \div 0.65) - [9 \times (1{,}150 - 965)]\} = 6.12$ *lb. per hp. hr.*

Example.—A bleeder turbine, which operates on saturated steam at 165 lb. per sq. in. gage, supplies a heating system which requires 12,000 lb. of steam per hour at 5 lb. per sq. in. gage. The turbine is rated at 1,000 hp. The condenser maintains a 29-in. vacuum at full load. What will be the total steam consumption of the turbine in pounds per hour at full load? Solution.—From Fig. 20, $\mathbf{E}_r = 0.60$. From Fig. 15, $\mathbf{H}_1 = 1{,}196$; $\mathbf{H}_2 = 835$; $\mathbf{H}'_1 = 1{,}196$; $\mathbf{H}_2' = 1{,}034$. From the given data, $\mathbf{W}_{B2} = 12{,}000 \div 1{,}000 = 12$ lb. per hp.-hr. Hence, by For. (30): $\mathbf{W}_{B1} = [1/(\mathbf{H}_1 - \mathbf{H}_2)]\ [(2{,}545/\mathbf{E}_r) - \mathbf{W}_{B2}(\mathbf{H}_1' - \mathbf{H}_2')] =$

$[1 \div (1{,}196 - 835)] \times \{(2{,}545 \div 0.60) - [12 \times (1{,}196 - 1{,}034)]\} = 6.37$ *lb. per hp. hr.* Hence, *the total steam consumption of the turbine* $= 6.37 \times 1{,}000 + 12{,}000 = 18{,}370$ *lb. per hr.*

191. To Compute The Steam Consumption Of A Bleeder Turbine At Any Load And Any Rate Of Bleeding when its consumption at various loads with no bleeding is known, use the graphs of Fig. 208 (Joseph Gershberg in *Power*, Oct. 11, 1921). It may be safely assumed that the economies of a bleeder turbine which is not bled are very nearly the same as those of a high-pressure condensing turbine of the same size and type. The diagram is limited in its application to turbines of 300 to 2,500 kw. capacity using steam at 125 to 150 lb. per sq. in. gage, bleeding at 0 to 20 lb. per sq. in. gage and condensing at 26 to 28½ in. of mercury.

EXPLANATION.—The fraction B/F_{100}, which is laid out on the horizontal scale of the diagram, is first calculated. B/F_{100} = (*the rate of bleeding of the steam, in pounds per hour*) ÷ (*the steam consumption of the turbine at full load—no bleeding—in pounds per hour*). This value is then found on the scale and followed vertically until the inclined-line graph is intersected which corresponds to the percentage of full load at which the consumption is to be calculated. The point of intersection is then projected and a value of the fraction F_B/F_C is read on the vertical scale. F_B/E_C = (*the consumption with bleeding at the rate B*) ÷ (*the consumption without bleeding at the same load*). The consumption without bleeding, multiplied by this F_B/F_C ratio, will give the consumption at the given rate of bleeding. See the following example.

EXAMPLE.—A turbine uses 10,000 lb. of steam per hour at full load and 6,000 lb. at half load, when there is no bleeding. What will be the consumption at ½ load when bleeding 5,000 lb. per hr.? SOLUTION.—Calculate $B/F_{100} = 5{,}000 \div 10{,}000 = 0.5$. Find 0.5 on the horizontal scale as indicated by the dotted line and trace up to where the 50-per cent.-load graph is intersected at A. Then move to the left and read the value of F_B/F_C, which is found to be 1.56. The consumption at half load with this rate of bleeding is then $6{,}000 \times 1.56 = 9{,}360$ *lb. per hr.*

192. Exhaust-steam Or Low-pressure Turbines Are Applicable under several conditions (see Sec. 35 for definition): (1) *To improve the economies of a condensing reciprocating-engine plant.* (2) *To utilize the exhaust steam from non-condensing reciprocating machinery.* (3) *As part of a compound unit, to run from the exhaust of a high-pressure turbine.*

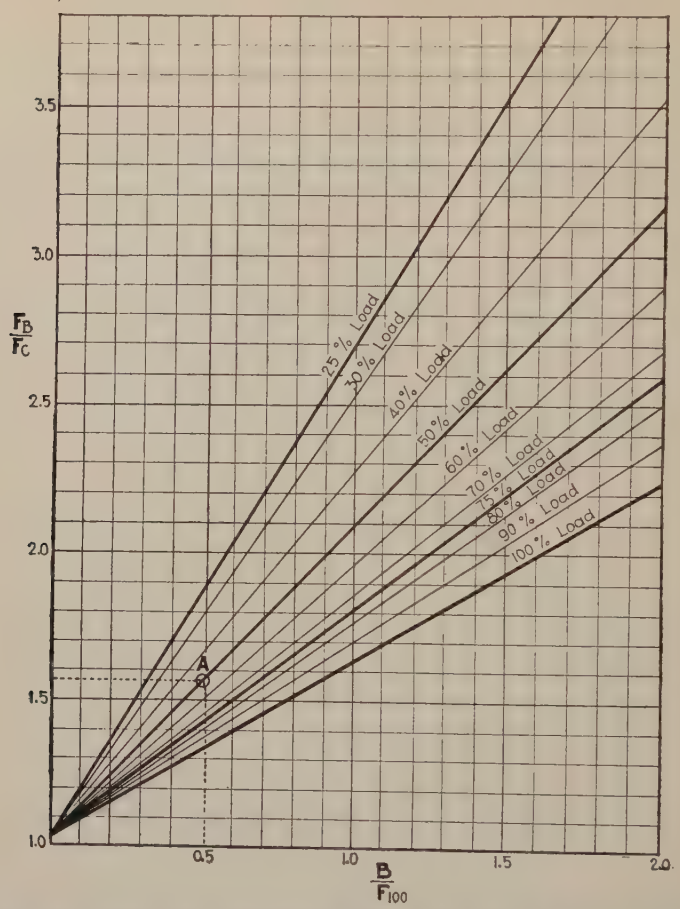


FIG. 208.—Diagram for estimating the steam consumption of bleeder turbines. For turbines 300 to 2,500 kw., 125 to 150 lb. steam pressure, 26 to 28½ in. vacuum, steam bled from 0 to 20 lb. gage pressure.

B = amount of steam bled in pounds per hour at any load.

F_{100} = amount of high-pressure steam required in pounds per hour at full load when no steam is bled.

F_B = Total amount of high-pressure steam required in pounds per hour when bleeding B pounds per hour at a specific load.

F_C = Total amount of high-pressure steam required in pounds per hour for the same specific load when no steam is bled.

Exhaust-steam turbines are usually either of the double-flow reaction (Fig. 79) or the Rateau type, although the single-flow reaction type (Fig. 209) is also used. The double-flow feature is used in the reaction type because of the large volume of steam which must be accommodated at the low pressure. The large volume is accommodated in turbines of the Rateau type by making the nozzle area proportionally large.

NOTE.—LOW-PRESSURE TURBINES ALWAYS OPERATE CONDENSING.—There is so little power available between the usual pressure of low-pressure steam (0 to 15 lb. per sq. in. gage) and atmospheric pressure that

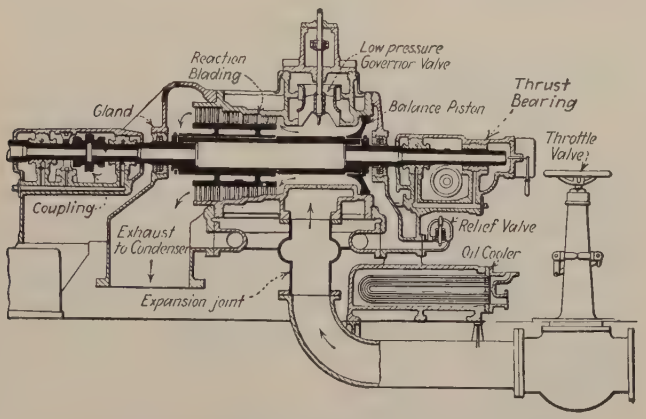


FIG. 209.—*Allis-Chalmers* single-flow low-pressure turbine.

no turbine would be justified for low-pressure non-condensing service.

NOTE.—IRREGULAR SUPPLIES OF EXHAUST STEAM CANNOT BE UTILIZED SATISFACTORILY BY A LOW-PRESSURE TURBINE ALONE.—When the supply of exhaust steam on which the turbine is to operate is irregular—as when the source is a steam hammer or a rolling mill engine—some means, such as a regenerator, of storing or accumulating a supply of this steam is sometimes used, (see Div. 8). Another method is to employ a mixed-pressure turbine; then the deficiency in exhaust steam is made up by drawing live steam from the high-pressure steam line.

193. The Addition Of A Low-pressure Turbine Usually Improves Both The Capacity And Economy Of An Existing Non-condensing Reciprocating-engine Installation (Fig. 210). The increase in capacity is usually 75 to 100 per cent. That is,

if the non-condensing engines develop 1,000 hp., the engine-and-turbine combination may develop 2,000 hp. The increase in economy, expressed as a percentage of the water rate is usually 30 to 50 per cent. That is if the engine operating non-condensing consumes 30-lb. steam per brake horsepower-hour, the engine-and-turbine combination may consume only 15 lb. per b.hp.-hr The first cost of a low-pressure turbine

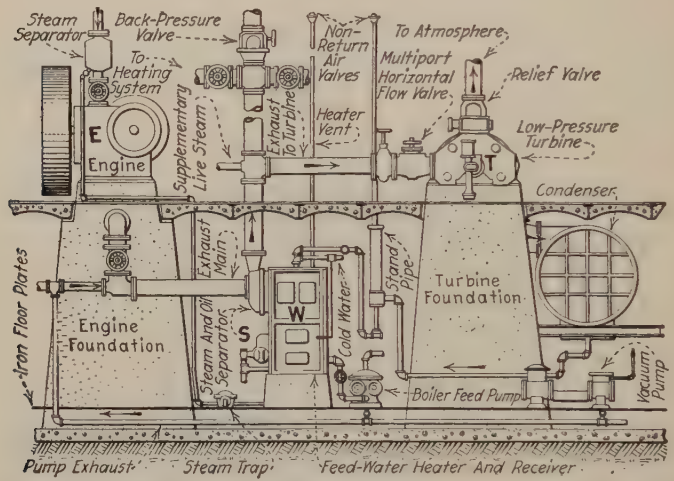


Fig. 210.—Low-pressure turbine, *T*, installed to operate on the exhaust from a non-condensing reciprocating engine, *E*.

is very low compared to the cost of an additional boiler and high-pressure unit for the same amount of additional power.

Note.—It Is Generally Well To So Arrange That Each Engine Will Supply Its Own Separate Low-pressure Turbine And Condenser, principally because, if one turbine and condenser served several engines, condenser or turbine trouble would render the entire outfit ineffective. Where there are a number of very small units, it may be better to provide but one turbine-and-condenser for a group of two or three engines to insure minimum first cost per kilowatt capacity. In any case, there should, preferably, be more than one complete low-pressure-turbine-condenser unit in each plant so that the danger of a complete breakdown will be a minimum. If several engines exhaust to one turbine-and-condenser, each engine should always in starting be run non-condensing a few strokes. This is to avoid impairing the condenser vacuum with the air which was in the engine cylinder when it was lying idle.

Note.—Receivers And Steam-and-oil Separators Should Ordinarily Be Installed Between Engines And Mixed- Or Low-pressure Turbines; see *S*, Fig. 210. The water and oil which is present in the engine exhaust may do comparatively little damage to the turbine if the oil is pure—except that they increase the friction of the turbine blading. But if the oil is impure and contaminated with matter taken mechanically from the boilers, it may form deposits on the turbine blades and thus seriously interfere with the operation of the turbine. A *receiver is usually necessary* to equalize the pulsations in the steam supply which result from the intermittent exhaust from the engine. In Fig. 210, the open feed-water heater, *W*, acts as a receiver.

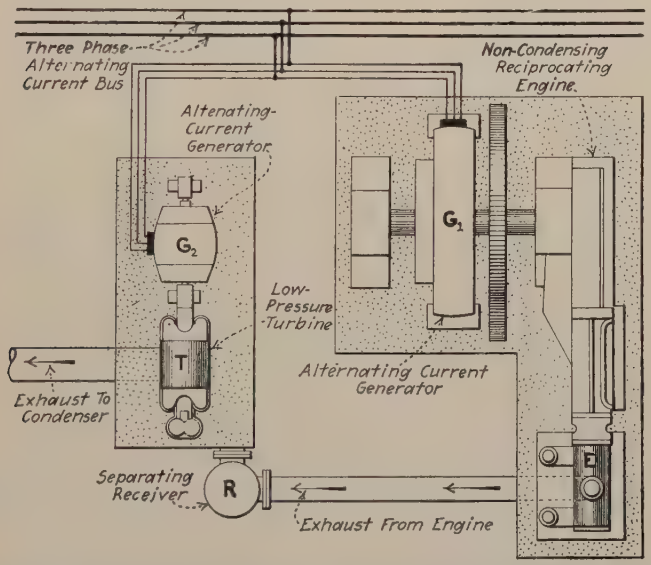


Fig. 211.—Diagram showing method of operating reciprocating-engine and low-pressure-turbine generating units on the same alternating-current line without, governing the turbine.

194. Several Methods Of Balancing The Load Between A Non-condensing Reciprocating Engine And A Low-pressure Turbine are shown in Figs. 211, 212, 213 and 214. It is desirable to have the engine in such installations produce exactly as much exhaust steam as the turbine requires. Then, all of the steam will be used with maximum economy.

Example 1.—When (Fig. 211) both the low-pressure turbine, *T*, and the non-condensing engine, *E*, drive alternating-current generators, G_1

and G_2, which are both connected to the same alternating-current line, the arrangement is self-balancing. The two units are automatically by electrical interaction between the generators, kept at the same synchronous speed. If their load characteristics are similar, the steam from the engine will always be just sufficient for the turbine—when once the loads have been balanced. There will not be any excessive variation of the exhaust steam pressure in receiver, R. The turbine may then be run without a speed governor. The engine governor and the turbine emergency governor serve to control the speed and provide protection.

Example 2.—When (Fig. 212) the engine drives a direct-current

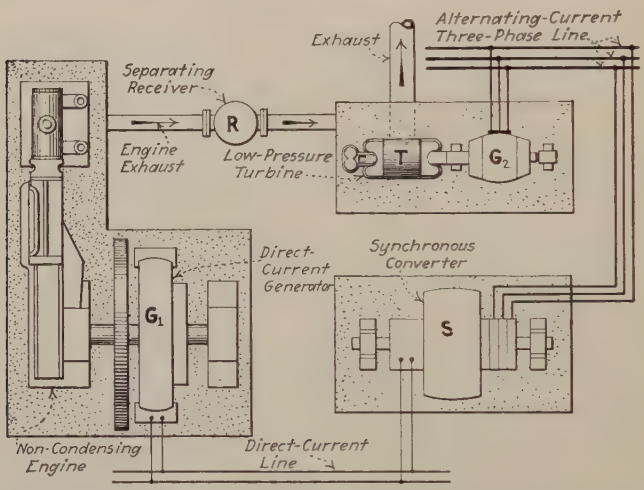


Fig. 212.—Method of supplying both alternating and direct current from turbine and reciprocating-engine generators, using synchronous converter for balancing the loads.

generator and the turbine an alternating-current generator or vice versa, the conditions are essentially the same as in Example 1 above except that a synchronous converter, S, must be employed to balance the alternating- and direct-current loads. There will be voltage fluctuations when the converter changes its function from maintaining the alternating-current voltage at the expense of the direct-current to the reverse operation but this fluctuation may be corrected at the switchboard.

Example 3.—Where (Fig. 213) the mechanical load, L_M, on the engine, E, and the electrical load, L_E, on the turbine, T, are balanced by means of a synchronous motor, M, the two units may be controlled by one governor as in Examples 1 and 2. Some adjustment at the switchboard is necessary when the motor changes over from acting as a motor to acting as a generator.

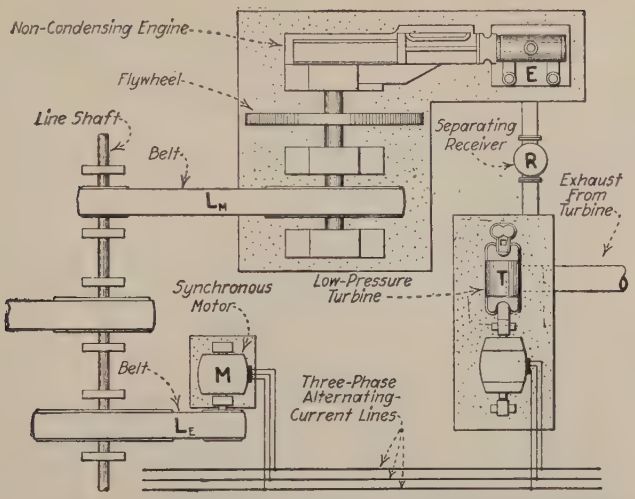


FIG. 213.—Showing how mechanical and electrical loads may be interconnected so that the power requirements of a mechanical-drive engine and the low-pressure turbo-generator which it supplies with steam will be balanced.

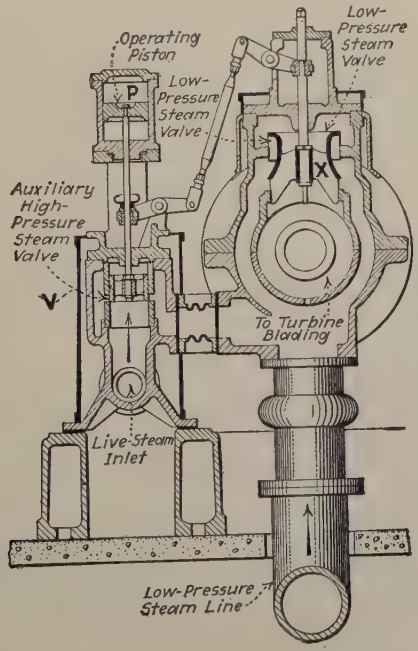


FIG. 214.—Transverse section showing the governing valve of a low-pressure turbine. Exhaust steam is admitted through valve *X* which is controlled as the operating piston *P* is actuated by oil from the governor relay valve. When *X* is wide open, further movement of *P* admits high-pressure steam through the valve *V*.

Example 4.—Where (Fig. 215) the two loads, L_1 and L_2, are entirely separate, both units, E and T, must be governed independently. The engine, E, will then furnish much more steam at times than the turbine, T, requires. The excess is automatically passed through the flow valve,

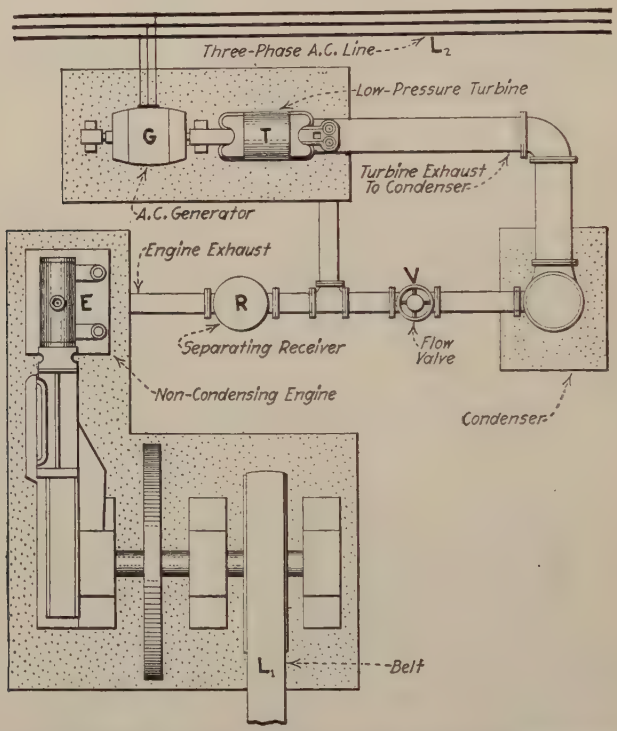


Fig. 215.—Application of a low-pressure turbine where a reciprocating engine drives a line shaft and always furnishes enough exhaust steam for the power requirements of the low-pressure turbine. The excess steam from the engine which is not needed by the turbine is condensed.

V (see Fig. 201), and is condensed. If the engine exhaust is occasionally insufficient for the turbine, a live-steam valve (V, Fig. 214) on the turbine will open and permit the deficiency to be made up; the low-pressure turbine then performs the function of a mixed-pressure turbine in a way but has, of course, no high-pressure blading.

QUESTIONS ON DIVISION 9

1. Name three special applications of steam turbines in power plants for which steam engines cannot be economically used. Two for which engines can also be used.

2. Why is a non-condensing turbine useful when much low-pressure steam is needed for heating?

3. Where no exhaust steam is available and none needed, what kind of turbine is ordinarily used?

4. If a condensing steam turbine develops 2,000 kw. on a given supply of steam, approximately how much power would a non-condensing turbine develop from the same steam supply under typical conditions?

5. Name two applications of a high-pressure non-condensing turbine. What types of turbines are preferred for each application?

6. A bleeder turbine combines the functions of what two other kinds of turbines?

7. What two devices are necessary for the governing of a bleeder turbine? What function must these two devices perform besides that of keeping the turbine speed constant?

8. A mixed-pressure turbine combines the functions of what two other kinds of turbines?

9. What are the functions of a mixed-pressure turbine governor?

10. What is the purpose of an automatic travel regulator for a mixed-pressure turbine governor?

11. What is the purpose of maintaining an automatic exhaust-steam heat balance in a power plant?

12. Show by a sketch how a mixed-pressure turbine may be connected to other power-plant equipment for maintaining an exhaust-steam heat balance.

13. Show by a sketch how a bleeder turbine may be connected to other equipment for maintaining an automatic exhaust-steam heat balance.

14. How do the costs of mixed pressure and bleeder turbines ordinarily compare with those of the other equipment which they can replace?

15. How do the speed regulations of mixed pressure and bleeder turbines ordinarily compare with those of other turbines?

16. On what two bases are the economies of mixed pressure and bleeder turbines considered? Explain how their steam consumptions may sometimes be practically zero on one basis.

17. When is a low-pressure turbine useful? Why is it sometimes economical to install one in a condensing reciprocating engine plant?

18. How much improvement in economy and capacity may usually be expected from the installation of a low-pressure turbine in a non-condensing reciprocating-engine plant?

19. What is the disadvantage of having all the engines in a plant exhaust to one low-pressure turbine and condenser?

20. Why are a steam separator and receiver advisable between an engine and a low-pressure turbine?

21. Show by a sketch how a low-pressure turbo-alternator is connected for parallel operation with an engine-driven alternator.

22. How may the load be balanced between an engine-driven direct-current generator and a low-pressure turbo-alternator? Explain with a sketch.

23. How may the load on a low-pressure turbo-alternator be balanced with that of an engine which is used for a line-shaft drive?

PROBLEMS ON DIVISION 9

1. In a power plant where the boilers deliver steam at 150 lb. per sq. in. gage and 50° F. superheat, the non-condensing steam engines consume 6,000 lb. of steam per hour and exhaust at a back pressure of 5 lb. per sq. in. gage. It is desired to utilize this exhaust steam in a 500-hp. mixed-pressure turbine which will exhaust into a vacuum of 28.5 in. of mercury column. About how much high-pressure steam will this turbine require per hour when operating at full load?

2. A 1,500-hp. bleeder turbine is to take steam at 180 lb. per sq. in. gage and 100° F. superheat. It will exhaust into a surface condenser where the vacuum will be maintained at 29 in. of mercury when the barometer stands at 30 in. It will also be required to supply 22,500 lb. of steam per hour for manufacturing purposes at a pressure of 10 lb. per sq. in. gage. Approximately how much steam will the turbine require from the boilers when it is operating under full load?

DIVISION 10

STEAM-TURBINE LUBRICATION

195. The Importance Of Steam-turbine Lubrication cannot be overemphasized because steam turbines operate at such high speeds and are constructed with such small clearances that a slight amount of wear may cause disastrous results. Perhaps no other phase of steam-turbine operation is more difficult and has given more trouble in the past than has lubrication. To secure satisfactory lubrication, three fundamental requirements must be observed: (1) *A suitable and high-grade oil must be used;* see Sec. 198. (2) *The oil must be properly supplied to the bearings;* Sec. 196. (3) *The purity and quality of the oil must be maintained;* Sec. 199.

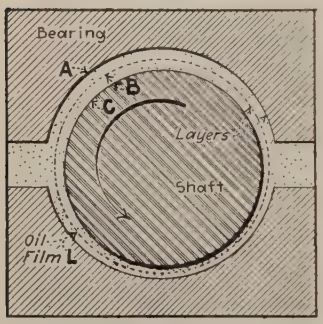


FIG. 216.—Showing how an oil film, *L*, maintains the position of a shaft in a bearing. The oil is assumed to divide into layers as, for example *AB* and *BC*.

NOTE.—THE FUNCTIONS OF AN OIL IN A BEARING are: (1) *To form a film between the journal and bearing,* Fig. 216, and thus to provide sliding between layers of the oil rather than between the metallic surfaces. See the author's STEAM-ENGINE PRINCIPLES AND PRACTICE for a discussion of the theory of lubrication. (2) *To carry from the bearing such heat* as is generated by friction in the bearing and as may flow to the bearing through the shaft. Sometimes, with ring-oiled bearings, water is circulated through the lower half of the bearing to assist in carrying away this heat, see Fig. 101.

196. The Methods Of Supplying Oil To Turbine Bearings are, briefly, two: (1) *Ring oiling,* Figs. 75 and 217 in which a ring (sometimes an endless chain) is supported on the journal and dips at its lower part into a small reservoir of oil in the

pedestal. As the shaft turns, it turns the ring which thus carries oil to the upper part of the journal whence it is carried,

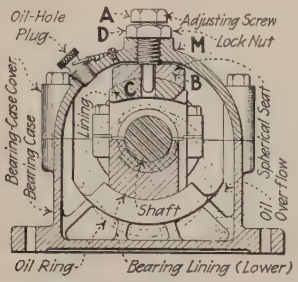


FIG. 217.—Bearing of the Type-6 *Sturtevant* steam turbine. There is an adjusting screw, *A*, in the bearing-casing cover, *M*. This screw when tightened down, causes the spherical seat, *B*, to grip the linings, *C*. The locknut, *D*, locks *A* in position.

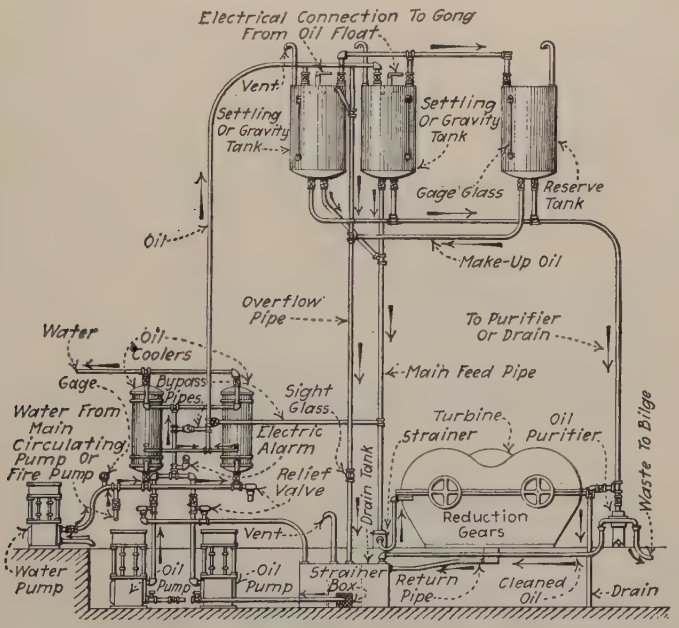


FIG. 218.—Gravity oiling system used on marine turbines. (*General Electric Company.*)

by the rotation of the journal, over the bearing surface. Cooling of the bearing is effected principally by radiation from the bearing and reservoir. Ring oiling is generally employed only

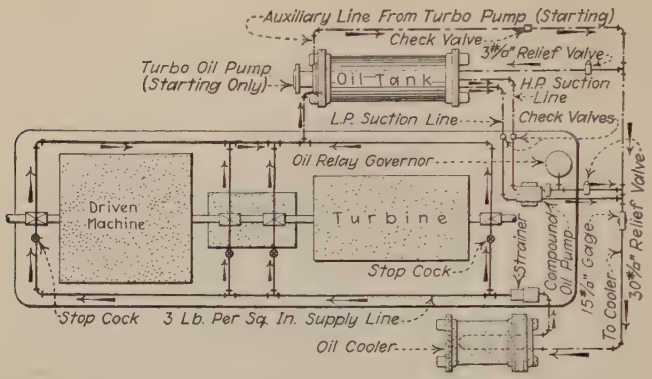


FIG. 219.—Piping diagram of the lubricating system of the *Kerr* turbine. The main oil pump is made in two parts—one to supply the governor, the other to supply the bearings. Hence, it is called a compound pump. The dash-dot lines indicate pipe lines which are below the floor.

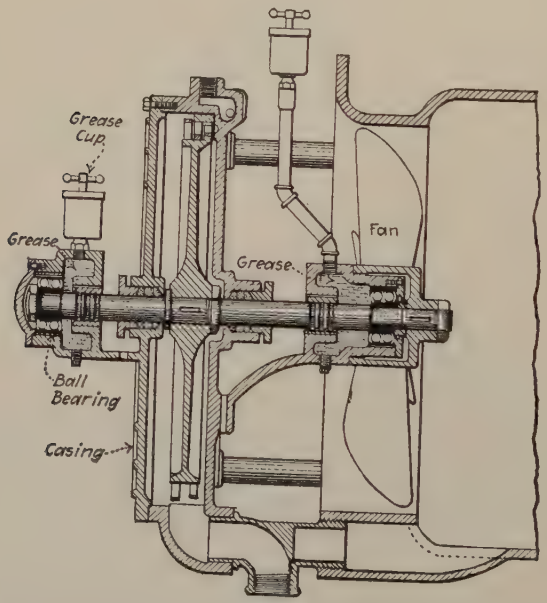


FIG. 220.—Cross-section through *Coppus* turbo-blower type *B*, showing grease lubrication of ball bearings. (The grease is forced down into the cavity beside the ball bearings by turning the handle on the grease caps to the right, thereby forcing down a plunger in the cup.)

on small turbines, up to about 300 hp. (2) *Circulation oiling*, Sec. 197, in which oil is conducted through pipes to and from the bearings. Circulation oiling is sometimes classified as, (1) *gravity circulation*, Fig. 218 and (2) *force-feed circulation*, Fig. 219, but these two classes differ only in the method of forcing the oil through the bearings—it flowing in one case by gravity, in the other it is forced by a pump—the actual oil pressure at the bearings being small in either case.

Note.—The Lubrication Of Ball Bearings is attained sometimes with grease which is supplied to them from compression cups (Fig. 220), and sometimes with oil which is supplied by rings as explained above (Fig. 50). Grease provides poor lubrication and should, generally, not be used except where a ball-bearing turbine is placed in a very dusty atmosphere—the grease then serves to keep impurities out of the bearings.

197. The Circulation System Of Turbine-bearing Lubrication, Fig. 221 is employed on nearly all turbines of sizes larger than 300 hp. and sometimes on smaller ones. The oil reservoir, D, and the cooler, C, are generally provided in the bedplate of the turbine. In the gravity systems the reducing valve, R, discharges into an overhead tank. The operation of the system is explained below.

Explanation.—In Normal Operation the main rotary oil pump, P, which is mounted on the turbine and driven from the turbine spindle, draws oil from the reservoir, D, through a strainer, S, and delivers it through the cooler, C, into a main feed pipe, M, at a pressure of 40 to 60 lb. per sq. in. gage. This pressure is generally required to operate the governor. A reducing valve, R, admits oil from M to N. In N the pressure is maintained at some value from 5 to 15 lb. per sq. in. gage by valves R and A which are adjustable for different pressures. If the pump supplies more oil than is needed by the governor and the bearings, the excess is bypassed through the relief valve B into D. The oil in N passes as shown, to the several bearings where it is admitted into grooves at or near the tops of the bearings and is drawn between the bearing surfaces. Oil vents, V, prevent the accumulation of air in pockets at the bearings and provide a convenient means for viewing whether a bearing is receiving sufficient oil. From the bearings, the used and excess oil flow as shown by the arrows back to D.

Should The Oil Pressure In M Fail because of clogging of the strainer, S, or for any other reason, the throttle valve would, in most turbines, be thereby automatically closed by the governor and the

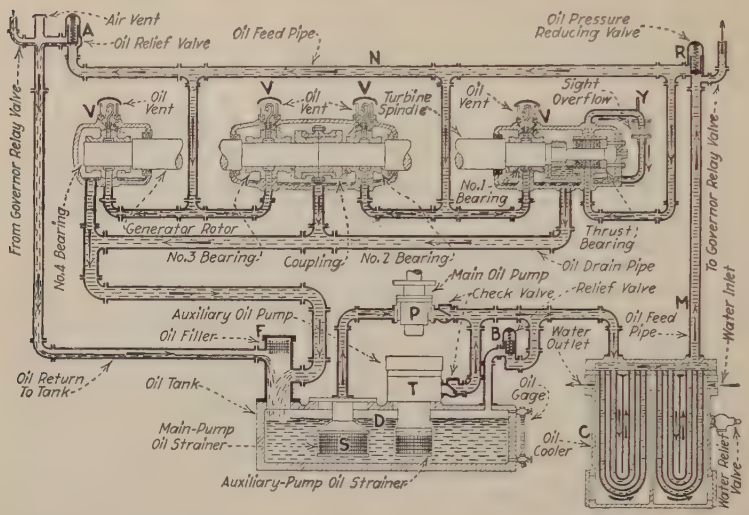


Fig. 221.—Diagram illustrating the flow of oil in a circulation oiling system. For bearing construction see Fig. 94. (*Allis-Chalmers Mfg. Co.*)

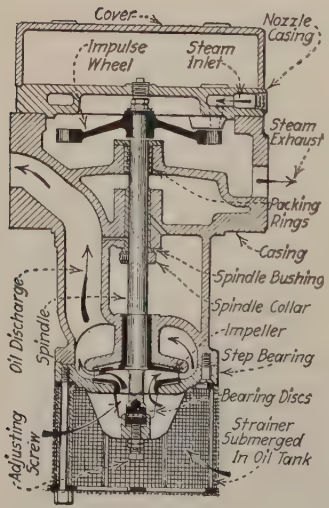


Fig. 222.—Turbine-driven auxiliary oil pump. (*Allis-Chalmers Mfg. Co.*)

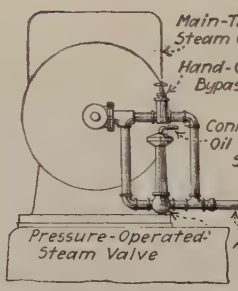


Fig. 223.—Pressure-operated valve for controlling steam supply to auxiliary-oil-pump turbine on *Allis-Chalmers* turbines. This valve automatically opens the steam supply to the auxiliary-oil-pump turbine and starts it whenever the pressure in the oil system falls below the normal value.

turbine would thus be stopped. Hence, *in starting the turbine the working oil pressure must be attained before the turbine can be supplied with steam.* For this reason, an auxiliary oil pump, *T*, (see also Fig. 222), driven by a small individual steam turbine, is supplied on each large turbine and is to be used in starting until the large turbine's speed is such that *P* can

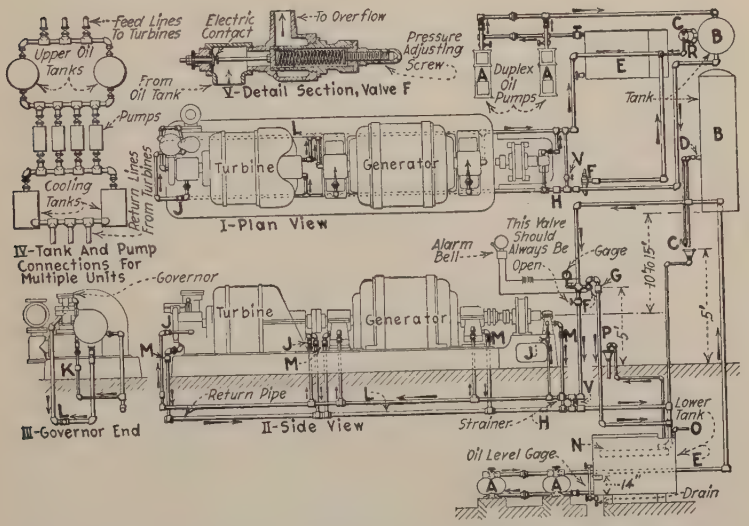


FIG. 224.—Oiling system of *Ridgway* turbines. Pumps *A* deliver oil into the overhead tank *B*. Valve *C* is left open until the oil level reaches *D*; then *C* is closed and the air above *D* is compressed. When the pressure in *B* exceeds that for which relief valve, *F*, is set (about 30 lb.), the oil flows through it and overflows at *G* into the lower tank, *E*. The oil which is not bypassed at *F* flows through the strainer, *H*, and thence through the feed-adjusting valves, *J*, to the bearings or through the strainer, *K*, to the governor. Sights, *M*, indicate the oil flow from the bearings into the return pipe, *L*. The used oil is filtered at *N*. Cooling water enters at *O* and leaves at *P*. A low oil pressure will allow *F* to close, which rings the alarm bell. If the oil pressure fails, the turbine should be stopped; the bearings will be supplied, while the rotor is stopping, by the oil in tank, *B*. The check valve, *R*, permits air to enter the tank in this event. The valve, *V*, should be opened only to drain the system.

supply sufficient oil. In the smaller turbines which are circulation-oiled and which do not employ oil-relay governors, oil rings are sometimes furnished to provide the necessary lubrication until the main pump attains a working speed.

Some manufacturers equip their auxiliary-pump turbines with a throttle valve which is automatically controlled by the oil pressure in the main pipe (Fig. 223). This prevents the main turbine from coming to

rest—which may take a half-hour or more—without a sufficient supply of oil in the bearings. Where the auxiliary-pump turbine is only hand-controlled, however, it should be started at the least indication of oil failure, and the main-turbine throttle valve should be closed as soon as possible.

Other manufacturers employ different schemes for supplying oil when the main pump fails. Figure 224 shows how an overhead tank may serve this purpose.

198. An Oil For Turbine-bearing Lubrication Must Possess Certain Properties; since the oil consumption of steam turbines is very low because the oil does not mix with or pass out with the steam or condensate but instead is largely used over and over again in a circulation system, it is always economical to use a high-grade mineral oil: (1) The *viscosity* should be such that the oil does not offer much resistance to dividing into layers—produce much friction—and yet the viscosity must be sufficiently high to insure an ample factor of safety against breaking down of the oil film in the bearing. An oil of high viscosity will cause excessive heating in the bearings and a consequent loss of power. Recommended viscosities are from 130 to 310 sec. Saybolt at 100° F., although viscosities above 200 sec. are seldom advisable; see the authors STEAM-ENGINE PRINCIPLES AND PRACTICE for method of measuring viscosity. Generally, the oil of the lowest viscosity (between the limits given above) that will give the desired oil pressure for the governor, should be used. (2) *Emulsification* should be small; that is the oil should separate rapidly from water when mixed with it. A good comparative test is to shake like quantities of two oils with water in a bottle and observe the rates at which they separate. (3) *It should be non-corrosive;* a piece of clean polished copper inserted for 5 hr. in the oil while the oil is kept in a bath of boiling water should show no darkening or diminution of the polish. (4) *It should have a flash-point* which is not below 325° F.; oils with lower flash points are likely to suffer a partial evaporation in the turbine bearings and gradually acquire a higher viscosity. (5) *It should not form deposits;* this property can, generally, only be determined after a trial of the oil.

NOTE.—EMULSIFYING AND CORROSIVE OILS ARE PARTICULARLY UNDESIRABLE FOR TURBINE-BEARING LUBRICATION because such oils are

almost certain to form a *sludge* or sticky compound which will clog the strainers, cooler tubes, and oil passages—thus impairing the lubrication and the cooling.

Note.—The Following Oils Are Recommended By Various Turbine Manufacturers: Vacuum Oil Company's *D.T.E. Light;* Texas Company's *Cetus;* Atlantic Refining Company's *Atlantic Turbine Oil, Light* or *Medium;* Sinclair Refining Company's *Cordymo;* Standard Oil Company's *Superla;* Gulf Refining Company's *Paramount Turbine Medium;* Tide Water Oil Company's *Turbol.* For turbines which are subject to excessive vibration or which use the same oil in reduction-gear and turbine bearings (see Sec. 203), a heavier grade should be used.

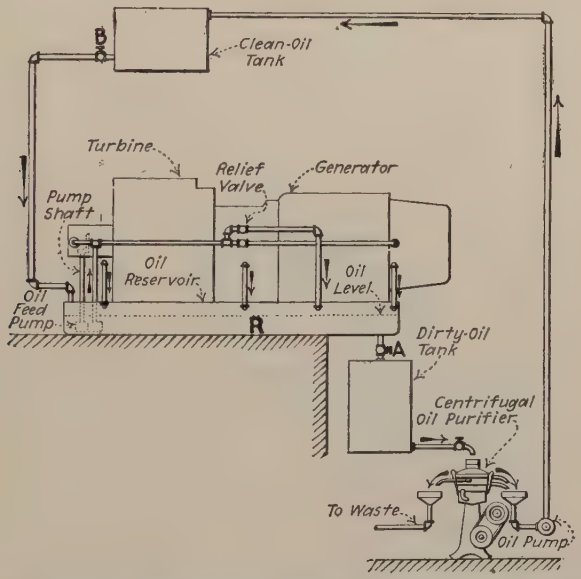


Fig. 225.—Arrangement of apparatus in a "batch" system of oil purification. The dirty oil is withdrawn through valve *A* into the dirty oil tank below the turbine. The valve *A* is then closed and the reservoir, *R*, cleaned. Then valve *B* is opened and a supply of clean oil flows from the upper tank to the reservoir. Valve *B* is then closed and the turbine is ready for operation. The dirty oil is passed through the purifier and is pumped back to the clean-oil supply tank. (*De Laval Separator Company.*)

199. The Practical Methods Of Maintaining The Purity And Quality Of The Oil Are: (1) *Make-up treatment,* wherein the oil is maintained by adding to that in the system, monthly or weekly, only as much oil as has been lost by leakage and evaporation. This treatment is satisfactory for ring-oiled bearings and is sometimes employed in circulation systems.

With this treatment, the oil should all be removed from the system every 3 to 6 mo. and replaced with fresh clean oil. If properly filtered, the oil may again be used in the bearings. (2) *Sweetening treatment*, wherein a small fraction of the oil in the system (3–6 gal.) is removed at regular intervals and replaced by fresh clean oil. During the intervals the oil which has been removed is thoroughly filtered and is later returned to the system. If sweetening is done daily, this treatment is very satisfactory. However, if the sweetening intervals are long or the amount of replenished oil too small the oil gradually

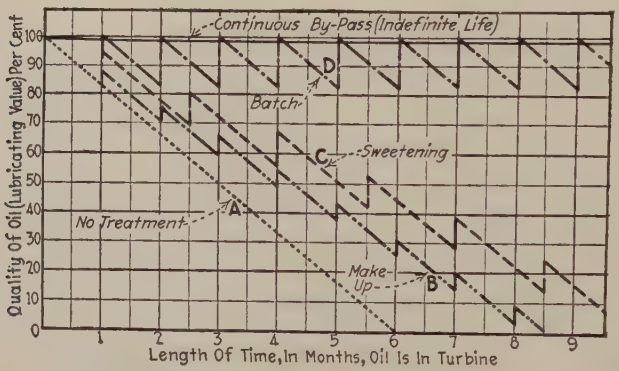


Fig. 226.—Graph showing effects of various methods of oil treatment. Graphs *B* and *C* might have different shapes and show better results if treatments are made with sufficient frequency. (*Richardson Phenix Co.*)

loses its lubricating value; see Fig. 226. (3) "*Batch*" *treatment*, (Fig. 225) wherein the entire oil supply is removed from the system at regular intervals and replaced with fresh clean oil. The oil which is removed is then filtered and purified—thus making it ready for replacement into the system at the end of the next interval. This method of treatment provides very satisfactory lubrication (Fig. 226) provided the intervals between treatments are not permitted to become too great; a month say, represents good practice. A disadvantage of this method is that the turbine must be shut down when the oil is replaced. (4) "*Continuous bypass*" *treatment*, Fig. 227, wherein a fraction of the oil in the system is continually passing through a filter, thus providing a continual "sweetening."

For turbines this method of treatment seems to be the best because it requires little attention and gives good results. (5) *Continuous treatment,* wherein the entire quantity of oil in the system is filtered each time it is handled by the main oil pump. Although this treatment is ideal, the necessary equipment is costly and requires much space. Hence it is seldom employed.

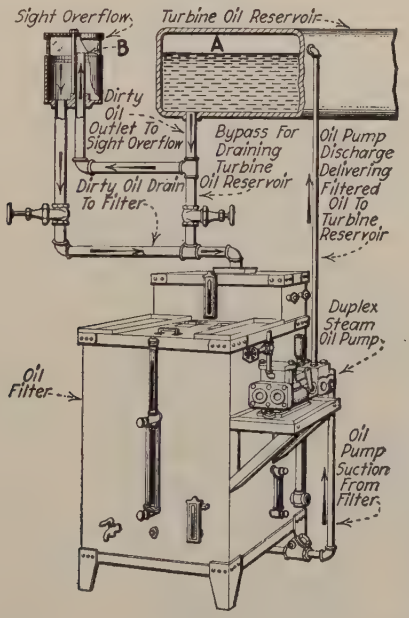

Sight Overflow
Turbine Oil Reservoir
B
A
Dirty Oil Outlet To Sight Overflow
Bypass For Draining Turbine Oil Reservoir
Oil Pump Discharge Delivering Filtered Oil To Turbine Reservoir
Dirty Oil Drain To Filter
Duplex Steam Oil Pump
Oil Filter
Oil Pump Suction From Filter

Fig. 227.—Illustrating one arrangement of apparatus for the "continuous bypass" system of oil treatment.

Note.—The Methods Of Purifying Oils are: (1) *Precipitation and filtration,* wherein the oil is heated, run slowly over trays, in which the water and heavier impurities settle out by gravity, and then is passed through cloth filter surfaces which remove the finer impurities. Many of the successful oil "filters," which are on the market, operate upon this principle. Their construction and operation are explained in the author's Steam-Engine Principles And Practice. (2) *Mechanical separation,* wherein the oil is separated from the water and heavier entrained particles in purifiers (Fig. 228) which operate on the principle of the well-known cream separator wherein centrifugal force is employed to effect the separation. Good results are reported with these purifiers.

They are made in different sizes to afford various capacities and require comparatively little power for their operation. (3) *Chemical purification* is generally necessary whenever an oil becomes acidified through use. A chemist should be supplied a sample of the oil and asked to recommend the proper treatment and, if possible, to determine the cause of the acidi-

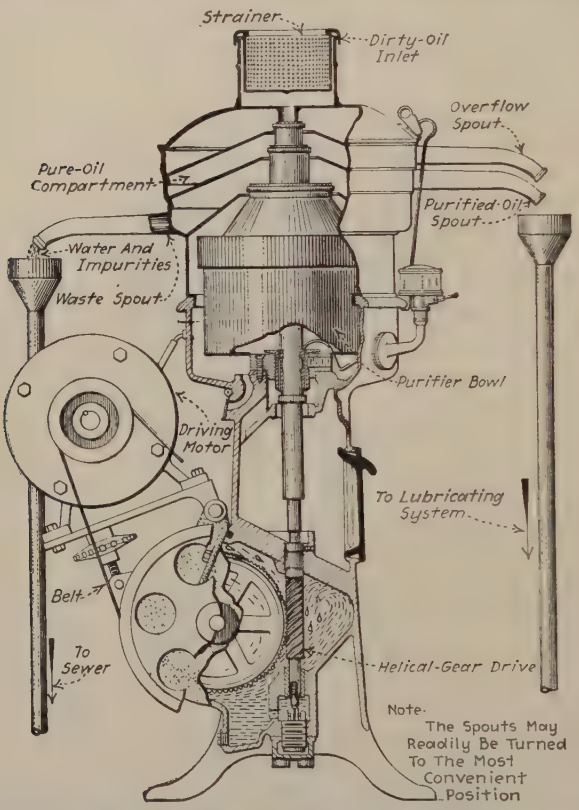


Fig. 228.—A motor-driven centrifugal oil purifier. These purifiers are also made for belt or steam-turbine drive. (*De Laval Separator Company.*)

fication. It should, in most cases, be possible to eliminate the trouble which started the acidification.

200. The Principal Causes Of Impurity Deposits In Oils are: (1) *Water.* Where considerable quantities of water leak into the system, emulsion takes place, and the oil takes on a yellowish color. Furthermore it is found that a *sludge* or a spongy

formation is evolved which, if permitted to remain in the system, will tend to clog the passages. The water generally leaks into the oil at the packing glands, Div. 5, or in the oil cooler. Water of condensation from a priming boiler wherein compounds are used and "hard" cooling water are particularly troublesome. (2) *Solid impurities*, such as fine particles of rust or moulders' sand, have a marked disintegrating effect on oil. Where they are present the oil assumes a dark color, and a "burnt" odor. A slimy dark deposit lodges on the surfaces, particularly in the cooler. Furthermore, in the presence of solid impurities, the oil will emulsify with very slight quantities of water which may collect in the system and will form sludge. (3) *Air* is usually present in the oil in greater or less amount and will, especially if the oil temperature is permitted to rise above normal—say 140° F.—tend to oxidize the oil. The oil darkens in color, increases in acidity, and in extreme cases a black carbonaceous deposit develops, which may choke the inlet to the bearings or cause sluggish movement of the governor gear or may even cause it to stick. (4) *Electric currents*, in some cases, may pass down through one bearing pedestal, through the bedplate, and up through the other pedestal—a portion of the current passes through the oil, darkens its color, increases its acidity, and throws down a deposit which coats all contact surfaces and lodges particularly in the cooler. The deposit is of a fairly hard, brittle nature and of a dark chocolate color; it is very difficult to remove. The remedy is to completely insulate one bearing from the bedplate; consult the turbine manufacturer. (5) *Adding new oil* sometimes causes deposits, especially where high-viscosity oils are employed.

201. Because One Function Of The Oil In Turbine Bearings Is To Carry Away Heat, The Oil Must Be Cooled, otherwise it would become too hot, lose its viscosity and become unsafe. Most of the heat is developed in the bearing by the friction between the layers of oil. Some heat also flows to the bearings from the steam inside the turbine casing.

Note.—Oil Coolers, *C*, (Figs. 221 and 229), are generally constructed of U-shaped copper tubes through which the oil (or water) is circulated while the outside of the tubes extends into the water space

(or oil reservoir). It would be preferable to have straight tubes as these are more easily cleaned. Although most manufacturers place the cooler in the turbine bedplate, it is better, if possible to have it separately mounted so that any vibration of the turbine would not be likely to produce leaks at the joints. The *pressure of the oil in the cooler should be greater than that of the water.* This will cause leakage to occur into the water rather than into the oil. The oil pump should, therefore, discharge through the cooler rather than draw oil through it by suction. Where only scale-forming (hard) water is available for cooling it may give trouble due to deposits on the tubes. In such event it is advisable to use the

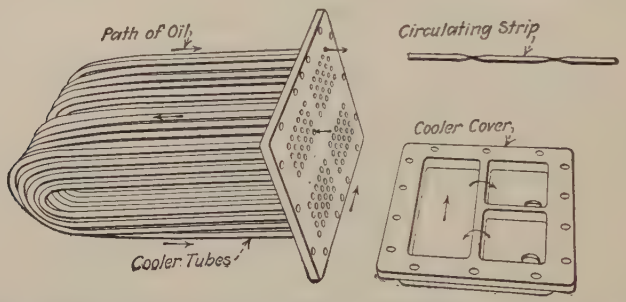


Fig. 229.—The essential parts of an oil cooler. Circulating strips, as shown, are placed in the tubes to give the oil a whirling motion through the tubes. (*Kerr Turbine Company.*)

same water over and over again by using a small cooling pond or tower to cool the water.

202. The Most Desirable Oil Temperatures For Bearings are: (1) *In circulation systems* the oil comes to the bearing at a temperature of about 100° F. and leaves at 130 to 140° F. However, no trouble is usually experienced if the oil enters at a higher temperature and leaves at a temperature not in excess of 160° F. Thermometers should be fitted to indicate the temperature of the water entering and leaving the cooler and, if possible, of the oil entering and leaving each bearing. These thermometers should be read once every hour and the temperatures recorded on an engine-room log sheet. (2) *In ring-oiled bearings* the temperature of the oil in the bearings, if of good grade, may safely be permitted to reach 200° F. or even a little higher temperature.

203. The Lubrication Of Geared Turbines (Fig. 169), since the service imposed on an oil in gear teeth is somewhat different

from that imposed in bearings, is a distinct problem. As long as the reduction gears are perfect and run noiselessly, the desirable bearing oil would also be satisfactory for their lubrication. But, should the gears become noisy, as they are likely to do, a heavier oil would then be needed in the gears. The heavier oil would, however, not be most desirable if the same oil is to be used in the turbine bearings because there it would almost certainly be contaminated with gland water which would not readily separate from the oil and would give trouble in the gears. For these reasons, *separate oil systems* should be provided for the turbine and for the gears.

204. The Lubrication Of A New Turbine Requires Special Attention because it is almost impossible to thoroughly clean the oiling system of all solid impurities. The impurities are very likely to cause deposits and hence cause trouble. The following procedure is therefore recommended for a new turbine.

Explanation.—Before starting the turbine all oil tanks, pipes, the cooler, and the like should be thoroughly cleaned to remove such solid particles as dust, grit, moulder's sand, rusty scale, and cotton waste. Cotton waste must never be used for cleaning oiled surfaces, as it leaves behind small particles which tend to clog the oil pipes and the small spaces in the governor mechanism. A smooth, lintless cloth or a sponge should preferably be used. The parts should be washed first with kerosene and finally with clean gasoline which should be wiped dry. The oil should then be poured into the reservoir—not directly but through the sieve—and the air should be expelled from the piping with the auxiliary oil pump.

After the turbine is started the strainers should be examined daily and, if necessary, cleaned. After a month's operation, the whole charge of oil should be removed from the system. The oil tanks, pipes, cooler and bearings should then be again thoroughly washed and cleaned. The system should then be filled with a complete charge of new oil. The oil which has been removed should be thoroughly purified and filtered before it is again put into the system. (It may be used as "make-up" oil.) This first change of oil may seem unnecessary but it will be found to pay in the long run; this is because a turbine requires the most care and attention in its early life. Later on, troubles should be rare if the oil is well looked after, frequently purified, and the strainers kept clean.

205. The Care And Operation Of A Steam-turbine Lubrication System—see also Sec. 204—involve: (1) *Attention to see that each bearing is receiving oil.* (2) *Observation of oil*

and water temperatures, as given in Sec. 202. Abnormal temperatures will readily disclose that something has gone wrong in the system and will usually give an indication as to the cause of the trouble. In case of abnormal temperatures the unit should be watched very carefully and shut down as soon as possible. Until the machine can be stopped, more oil should be fed to the bearings by increasing the discharge pressure on the pump or by starting the auxiliary pump if necessary. (3) *Regular treatment of the oil,* according to the method (Sec. 199) which is employed. With regular and proper treatment a good turbine oil should have a life, under favorable conditions, of 10,000 working hours or more, or of 3,000 working hours under very unfavorable conditions.

Note.—The Signs Of "Breaking Down" Of An Oil are: (1) *Darkening in color.* (2) *Increased specific gravity.* (3) *Increased viscosity.* (4) *Increased acidity.* (5) *The throwing down of various kinds of deposits.* Although all oils are affected in time, unsuitable oils will break down much sooner than will suitable oils. The best oil for a system is therefore the one which will last the longest without breaking down.

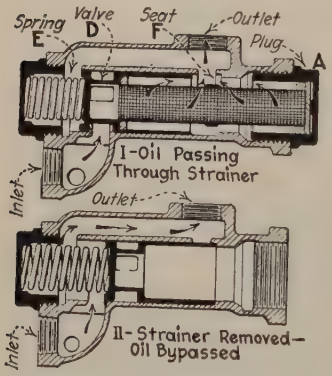


Fig. 230.—Oil strainer which is used on *General Electric Company's* 500-kw. turbo-alternators. View *I* shows the normal operation. To clean the strainer, nut *A* is unscrewed. Spring *E* then forces the valve *D* against the seat *F* thus permitting the oil to flow directly to the outlet as shown in *II*. After cleaning, the strainer can be replaced in like manner.

206. Some Useful Operating Hints On Steam-turbine Lubrication are given below:

Do Not Pour Oil Into The Reservoir Except Through The Strainer.—It saves time in the long run to pour it through the strainer. Furthermore, since the strainer had to be *bought,* make it pay for itself.

Take Out The Strainers (Figs. 230 and 231) And Clean Them Often. If the strainer is on the pressure side of the oil pump see that the oil is bypassed around the strainer before removing it.

Remember That Nearly All Turbines Have Some Parts Which Require Hand Oiling.—See that these parts get a few drops of oil *every day.* Also *keep the oil cups or drop-feed oilers filled.*

The Proper Oil-level For A Ring-oiled Bearing is generally indicated by a scratch on the oil gage glass. See that the oil level is maintained.

Watch Ring-oiled Bearings To See That The Rings Revolve.—Sometimes a ring will wear eccentric and fail to supply oil.

Pressure Gages On The Oil System Should Be Throttled so that very little or no vibration is visible. A vibrating gage wears rapidly.

Try To Have The Water And Oil Flow Through The Cooler In Opposite Directions ("Counter-flow").—In this way less water is needed to cool the oil than otherwise.

A Convenient Way To Clean The Oil Tubes Of A Cooler is to first blow them out with compressed air, then push through a flexible wire, fasten a clean cloth to one end of the wire, and pull the cloth through the tube. If this does not remove all

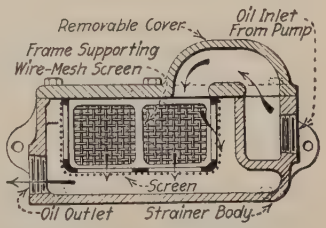


Fig. 231.—Section through oil strainer used on *Kerr* turbines.

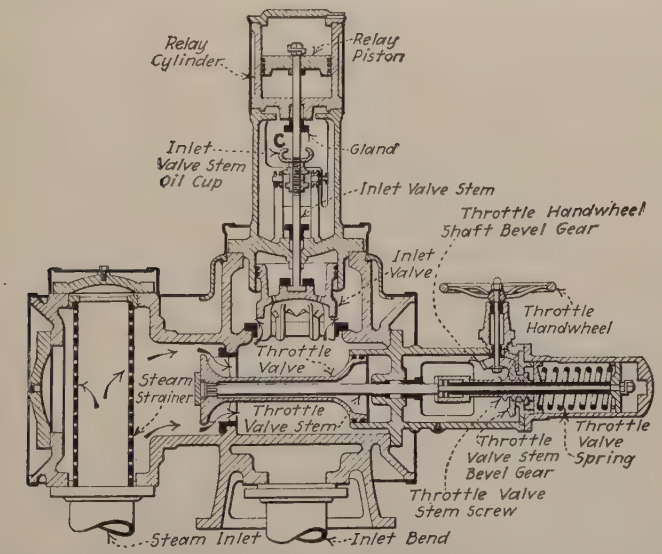


Fig. 232.—Section through throttle valve and steam chest of Allis-Chalmers turbine showing the oil cup, *C*, on the governor-operated inlet-valve stem. This cup collects oil which may leak through the gland from the relay cylinder and thus prevents this oil from "baking" on the hot valve stem.

deposits wrap the cloth with a brass gauze and pull through again. This will clean the tubes thoroughly.

Provide Oil Shields, where the governor relay cylinder is located above the throttle valve (Fig. 232), to prevent oil which may leak out of the cylinder from flowing to the throttle-valve stem. If not prevented, the oil will "bake" on and impair the operation of the throttle valve.

Let The Price Of An Oil Be Your Last Consideration in making a selection. A "cheap" oil is expensive in the long run.

Always Draw Off From The Bottom Of The Reservoir And, If Possible, After The Turbine Has Stood Idle A Few Hours.—The water and impurities will thus be removed.

Take Oil Samples From The Reservoir Once A Week.—A 4-oz. bottle should be filled, labeled, and placed away in a safe place. A comparison of these samples will often reveal troubles.

QUESTIONS ON DIVISION 10

1. Why is the lubrication of steam-turbine bearings of such vital importance? What three requirements are fundamental in steam-turbine lubrication?

2. What are the two functions of an oil in steam-turbine bearings?

3. Describe the ring-oiled method of bearing lubrication. What kind of turbines are most generally ring oiled?

4. What are the two principal classes of circulation-oiling systems? Wherein do they differ?

5. How are ball bearings generally lubricated? Show with a sketch how to keep grit out of ball bearings.

6. Describe fully, using a diagrammatic sketch, the operation of a circulation oiling system. What provisions are made, in circulation systems, for supplying oil to the bearings in case the main oil pump should fail?

7. State the five principal properties which an oil must possess if it is to be satisfactory for turbine-bearing lubrication and tell the reason for each property.

8. What kinds of oils are particularly undesirable for turbine lubrication? Why?

9. State the five methods of maintaining the purity and quality of an oil, describe each fully, and where possible draw a sketch of the apparatus required.

10. Describe the three methods of purifying oils and give the usefulness of each.

11. What are the five principal causes of deposit formations in oils? Explain the term *sludge*.

12. Discuss the construction and operation of oil coolers. How can the leakage of water into the oil be most easily prevented?

13. What are the desirable and permissible working temperatures of turbine-bearing oils?

14. Explain fully the distinctive features of geared-turbine lubrication. What method of lubrication is best adapted?

15. Why is the lubrication of a new turbine such an important matter? State what procedure and what precautions should be exercised.

16. What are the three important phases of the care of a steam-turbine lubrication system during operation?

17. What physical signs indicate that an oil is losing its lubricating value?

18. State a number of lubrication "pointers" which should be observed in operating a steam turbine.

DIVISION 11

STEAM-TURBINE OPERATION AND MAINTENANCE

207. The Three Fundamentals Of Steam-turbine Operation are, in the order of their importance: (1) *Safety*. (2) *Service*. (3) *Economy*. In other words, the operator should, above all, endeavor to make the operation of a turbine as nearly free from the possibility of accident as he reasonably can; his next consideration should be toward eliminating the likelihood of a necessary shut-down; then, after these first two elements have been attended to, he should aim to so operate the machine that the economy of the plant in its use of steam is the best that can be attained. Safety should never be sacrificed for the sake of service or economy. Operating methods which will tend to comply with the above fundamentals are given in following sections.

NOTE.—SOME GENERAL PRECAUTIONS SHOULD BE OBSERVED IN OPERATING STEAM TURBINES.—The most important ones are given below. These precautions must be taken seriously to heart if one desires to obtain satisfactory operation of the turbines under his care.

1. UNDERSTAND YOUR TURBINE PERFECTLY.—The preceding divisions were intended to familiarize the reader with the principles and usual construction of turbines of various types and their parts. Make sure that you also have the manufacturers' instructions for the turbine which you are to operate. *Read them carefully* and be sure that you understand them. Watch or supervise the installation of the turbine and be certain that you know the purpose of every piece, bolt, or nut. Know what is inside and out. The reasons that manufacturers have for doing certain things in certain ways may not always be apparent, but it is safe to assume that each piece has a purpose, and that the directions which they give have a sound basis. If a man is sent from the factory to acquaint you with the turbine (as is usually done with large turbines) ask lots of questions—he will be glad to answer them. If no man is sent, or if he is already gone when a question arises, write to the factory—it may save your life. It should always be remembered that the builders of the turbine know more than anyone else about the way in which that particular turbine should be operated.

2. Be Sure That Everyone Concerned With The Care And Operation Of The Turbine Understands It Perfectly.—If the turbine room must be left to someone else, be sure that he is competent. Don't be afraid that he will "get onto" your business. Remember that you will get the blame if anything goes wrong.

3. Don't Think That All Turbines Are Alike.—The fact that you understand one turbine does not signify that you are competent to go into another plant and immediately take charge of different turbines even if they are made by the same builder. Every turbine has its own peculiarities which must be determined by careful study.

4. Do Not Change The Operating Conditions.—Steam pressure, superheat and vacuum—without first consulting your instruction book and, if the point is not covered there, writing to the manufacturer. If it is necessary or if you deem it advisable to change the steam pressure, back pressure or vacuum, extract steam, or admit steam at mixed pressures, be sure that you know what the manufacturer has to say about such a change. There may be small but vital details of such operation which you would not think of, or it may not be advisable to make such a change—but the manufacturer will know and will be glad to advise you.

208. To Insure Safety In Steam-turbine Operation it is necessary always to observe the following points: (1) *Be sure that the main governor operates satisfactorily.* Whenever possible examine its parts for wear, lost motion, and sticking. When the unit is shut down see that the governor valve or its seat are not worn so that it cannot shut off the steam. (2) *Be sure that the emergency governor operates satisfactorily.* At least once every week or two the turbine should be speeded up to 10 or 15 per cent. over its rated speed (according to the manufacturers instructions) to insure that the emergency governor is in good order. Also, the turbine should always be shut down by tripping the emergency. (3) *Keep a careful watch of the turbine,* examining it every hour for oil temperatures, hot bearings and vibration (Sec. 212). (4) *Be sure that the automatic vacuum breaker operates satisfactorily,* if one is in the equipment. It is well to have a hand-operated vacuum breaker located near the throttle valve so that, if the automatic valve fails, the vacuum can be quickly broken by hand. (5) *Be sure that the atmospheric-exhaust valve works properly* and does not stick.

209. To Insure Uninterrupted Service In Steam-turbine Operation the following attention is quite essential: (1) *Provide*

adequate and proper lubrication, see Div. 10. (2) *Always have a spare unit ready*, if possible, to start on a moment's notice. When a main or spare unit is shut down for inspection or repairs, see that the work is done as quickly as possible so that it will be available in case something goes wrong. The repairs should be so planned that any interruption of service due to forced shut-downs will be a minimum. (3) *Make an inspection of the complete unit and auxiliaries at least once a year.* The unit should be completely dismantled and every part inspected for wear and cleaned. In reassembling, the worn parts should be carefully adjusted—or even replaced where necessary.

210. To Insure Maximum Economy In Steam-turbine Operation, try to: (1) *Maintain the nameplate steam pressure and superheat*, see Div. 13; this can be done by cooperating with the boiler-room force. (2) *Maintain the nameplate vacuum*, see Div. 13; the condenser may need frequent attention to see that the tubes or jet nozzles are not fouled and that air is not leaking in. (3) *Maintain the nameplate speed;* remember that turbines are designed to operate with the best economy at their rated speed. (4) *Operate the turbine at its most economical load*, if possible. If more than one turbine must be operated to carry the total load it is sometimes best to have some machines run at their most economical loads and one to take the fluctuations whereas sometimes it is best to run all the machines somewhat below their most economical load. The most economical arrangement should be determined by test or by reference to the individual performances of the several turbines and this arrangement should then be followed.

211. The Principal Troubles Which Arise In Steam-turbine Operation and which must be guarded against are: (1) *Unequal expansion of different parts during starting;* see method of starting in Secs. 213 to 215. (2) *Water in the casing;* slugs of water may be prevented from entering the casing by making the piping free of pockets and employing a separator ahead of the throttle valve. (3) *Overspeeding;* this is guarded against by periodic inspection and tests of the governor and overspeed tripping device. (4) *Excessive pressure in the casing;* this is prevented by the atmospheric-relief valve, which should there-

fore be kept in satisfactory condition. (5) *Vibration;* see Table 212. It is a peculiar fact that nearly all troubles which are experienced with steam-turbines—excepting (3) and (4) above— will manifest themselves sooner or later as vibrations. Hence, the chief duties of a turbine operator, while a turbine is in operation, are to carefully guard against overspeeding, excessive pressure, and vibrations.

212. Table Of Vibration Causes, Remedies, and Identifying Symptoms.

(Adapted from E. V. Amy, in *Electric World*, vol. 74, p. 1004)

Cause	How identified	Probable reason	What to do
Unbalance	Uniform vibration throughout machine; same frequency as speed; becomes slightly less as load is applied; intensity of the vibration depends on amount of unbalance.	(*a*) Sprung shaft. (*b*) Improperly placed balance weights. (*c*) Displacement of balance weights. (*d*) Sediment in blades or buckets. (*e*) Corroded blades. (*f*) Unequal heating of rotor parts. (*g*) Unbalanced forces due to heavy distortional stresses. (*h*) Shifting of conductors on generator. (*i*) Unequal generator air gaps.	As soon as vibration becomes abnormal, shut down and investigate. Remove cause; rebalance if necessary.
Poor alignment	Vibration of variable periodicity; slight at no load, becoming worse as load is applied.	(*a*) Eccentric coupling. (*b*) Unequal settling of foundation. (*c*) Steam-piping strains due to expansion or weight.	Check up and correct.
Bad foundation	Sympathetic vibration in surrounding structure; vibration felt all over machine and constant for all loads.	(*a*) Improper grouting. (*b*) Non-rigid fastening of bedplate. (*c*) Non-homogeneous foundation resulting in unequal settling.	Make foundation solid; grout with lead if necessary.
Loose parts	Vibration of local nature; greatest at loose bearing; rattle or noise when starting or slowing down.	(*a*) Too much bearing clearance. (*b*) Ball joint of bearing loose. (*c*) Loose construction in built-up rotor. (*d*) Loose conductors on generator. (*e*) Loose coupling or bolts.	Carefully examine all bearings and fastenings. Make necessary repairs.
Internal rubbing	Abnormal vibration somewhat localized; noise varying with speed of machine.	(*a*) Revolving buckets coming in contact with stationary buckets. (*b*) Insufficient casing clearance. (*c*) Deflection of a diaphragm or disc in one stage. (*d*) Thrust-bearing troubles.	Make repairs or adjustments immediately; may result in serious damage.

Cause	How identified	Probable reason	What to do
Steam troubles	Unusual noise near the intake; failure of the steam strainer.	(*a*) Water coming over with the steam. (*b*) Sediment in the steam. (*c*) Faulty valve gear causing irregular steam admission. (*d*) Accidental closing of emergency steam valve shifting generator's load to other machines.	Make necessary repairs; test steam for sediment, acid, or salt.
Packing troubles	Local vibration; noise; heating of shaft or packing casing.	(*a*) Improper adjustment of labyrinth packing. (*b*) Packing rings too small for shaft.	Make adjustment or replace old packing.
Oil troubles	Heating of bearing; noisy operation; may cause damage to blading due to lowering of spindle.	(*a*) Breaking down of oil film due to insufficient supply. (*b*) Oil supply cut off or too slow. (*c*) Poor oil (frothing, gumming, emulsifying).	Improve oil system; keep clean and well filtered.

213. In Starting A Newly Installed Turbine For The First Time, especially great care should be exercised. The regular starting operations, as described in the following sections, are to be followed but they should be very cautiously and slowly pursued. During the entire starting period there should be a constant watch for scant oil flow, heated bearings, blade rubbing, vibration trouble, and any extraordinary happening. If any trouble is noted, the turbine should be immediately stopped, the trouble corrected if possible, and then the turbine should be started all over again. If vibration is experienced, try to ascertain the cause and correct it. If a slight vibration begins at a certain turbine speed each time that the unit is started and if no cause can be found, try then to increase the speed and thus ascertain if the vibration will cease at a higher speed; the speed at which vibration begins may be a critical speed (Sec. 89) of the rotor shaft. Do not, however, try to pass any speed at which excessive vibration occurs. In case the cause of some trouble cannot be determined, call on the manufacturer for an expert erecting engineer—it is better to do this than to wreck your plant. It is best to permit the initial starting of a turbine to consume several hours and then to apply the load very gradually so that the machine can gradually "wear itself in." It is always advisable, and will doubtless save money in the end, to engage an erecting

engineer from the turbine manufacturer to erect and start all moderate- and large-capacity turbines.

214. To Start A Non-condensing Turbine (Fig. 233), follow the manufacturer's operating instructions. It is impossible to here give any set of directions which will apply to all turbines. The following procedure, will, however, apply in nearly all cases.

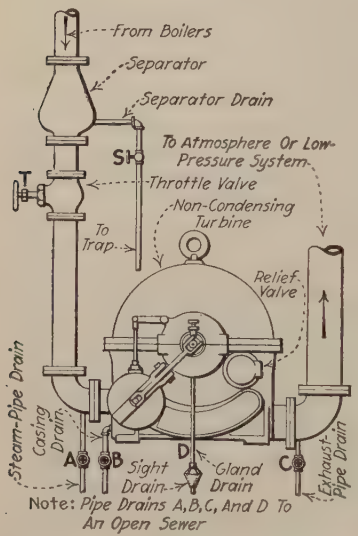


Fig. 233.—The principal steam and drain piping and valves of a non-condensing steam turbine. All drain pipes should lead from the lowest point of the chambers which they are to drain.

Procedure.—(1) Start the auxiliary oil pump, if the turbine has one. If the auxiliary oil pump is turbine driven, first start the pump turbine by following instructions 2 to 10 below. When the oil pump is delivering the required pressure, proceed with the starting of the main turbines as directed below.

2. Open all valves in the drain pipes from the steam piping and turbine casing; for example, valves *S*, *A*, *B*, and *C*, Fig. 233.

3. Inspect piping to see that the exhaust is clear and see that there is ample oil flowing to all bearings.

4. Open the throttle valve, *T*, Fig. 233, *quickly* but just enough to start the rotor spinning.

5. Immediately, as soon as the rotor starts turning, trip the automatic overspeed valve by operating the hand trip lever (not shown in Fig. 233). This is to insure that the overspeed valve is not sticking and that it shuts off the steam. See that the rotor comes to rest.

6. Reset the emergency overspeed valve.

7. Again open the throttle valve, *T*, Fig. 233, to start the rotor and adjust the valve to give a turbine speed of about 200 r.p.m. Let the rotor turn at this speed long enough to insure that the turbine is thoroughly warmed (3 or 4 min. on small turbines to 10 or 15 min. on large ones).

8. See that all bearings are receiving the proper amount of oil or that the oil rings are turning on the shaft.

9. Start water flow through the cooler and bearings (if water-cooled).

10. *Gradually* open the throttle valve, *T*, Fig. 233, to increase the speed of the turbine. See that, at the proper speed, the governor takes con-

trol. Then open T to its limit and close it one-half turn to prevent it from locking open.

11. Shut down the auxiliary oil pump and see that the main pump keeps up the pressure.

12. Close the valves in the drain pipes (A, B, and C, Fig. 233). If wet steam is used by the turbine, the drains should be left "cracked."

13. Apply the load to the turbine *gradually;* see Sec. 219.

NOTE.—THE ROTOR SHOULD BE SPINNING WHEN IT IS BEING WARMED.—This is very *important.* If less steam is admitted to the casing than is sufficient to turn the rotor, the steam will flow through the casing at the top, heat the upper part of the rotor and casing, and thus cause unequal expansion of the rotor and casing. Later, then, when the rotor is permitted to turn, the distorted rotor is very likely to cause rubbing of the blades or a sprung spindle. But, by allowing the rotor to turn slowly while starting, it is warmed evenly on all sides and the cold air is quickly drawn from the casing. Thus unequal expansion is prevented.

215. To Start A Condensing Turbine (Fig. 234), follow the manufacturer's operating instructions. It is impossible here to give any set of rules which will apply to all turbines. Some manufacturers recommend starting their turbines under full vacuum, some under a partial vacuum (24 to 26 in.) and some recommend starting under non-condensing conditions. Whichever method is recommended by the manufacturer should be followed. The following procedure will be satisfactory in most cases.

PROCEDURE.—1. Start water flow through the oil cooler and bearing cooling-coils and be sure that there is sufficient oil in the system.

2. Open all drains—valves S, A, G, and D, Fig. 234—and the turbine stop valve X.

3. Start the condenser pumps; W, C, and V, Fig. 234. The dry vacuum or air pump may be run slowly so as to produce no vacuum in the condenser. Turbines with steam sealed glands may be started condensing by opening the sealing valves, K, E, and F, Fig. 234, but the vacuum during starting should not exceed about 25 in.

4. Start the auxiliary oil pumps and adjust oil flow to all bearings.

5. Open the throttle valve, T, Fig. 234, *quickly* to start the rotor into motion.

6. Immediately, as soon as the rotor starts turning, trip the automatic overspeed valve by operating the hand trip lever (not shown in Fig. 234). This is to insure that the overspeed valve is not sticking and that it shuts off the steam. See that the rotor comes to rest.

7. Reset the emergency overspeed valve.

8. Again open the throttle valve, *T* (Fig. 234), to start the rotor and so adjust the valve that the rotor turns at about 200 r.p.m. Let the rotor turn at this speed long enough to insure that the turbine is thoroughly warmed (about 1 minute per 1,000 kw. of turbine capacity but in no case less than 10 minutes).

9. See that all bearings are receiving the proper amount of oil or that

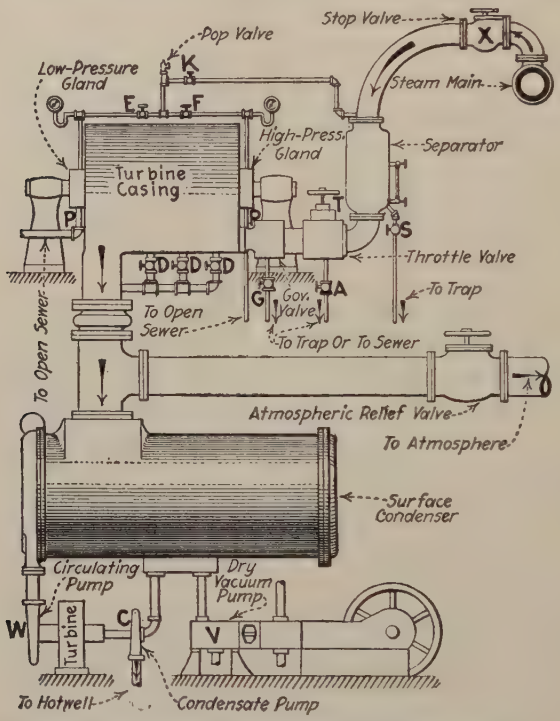


FIG. 234.—Typical arrangement of piping, valves and auxiliaries for a condensing turbine.

the oil rings are turning on the shaft.

10. *Gradually* open the throttle valve, *T* (Fig. 234), to increase the speed of the turbine. See that the governor takes control at the proper speed. Then open *T* to its limit and close it one-half turn to prevent it from locking open.

11. Water sealing glands may now be put into effect by turning on the water gradually. Then the vacuum may be raised to about 25 in. of mercury.

12. Shut down the auxiliary oil pump and see that the main pump keeps up the pressure.

13. Close the valves in the drain pipes (*A*, *G*, and *D*, Fig. 234). If wet steam is used by the turbine, the drains should be left "cracked."

14. Apply the load to the turbine *gradually;* see Sec. 219.

15. Build up the proper vacuum by regulating the condenser pumps.

16. If a steam seal is used on the glands, close the valve—*F*, Fig. 234—in the pipe leading to the high-pressure gland.

NOTE.—TO START A BLEEDER OR A MIXED-PRESSURE TURBINE, close the low-pressure steam valve and start as directed above for a condensing turbine. After the turbine is running under full load, *gradually* open the low-pressure valve.

216. The Care Of A Turbine While It Is Running involves only a periodic (about hourly is generally sufficient) inspection

DAILY LOG SHEET FOR Main TURBINE No. 3 Date 4/27/22

	Time	STEAM PRESSURES Supply	Stage 1	Stage 2	Stage 3	OIL PRESS. Relay	BEARING TEMPERATURE Bearing	Inlet	Outlet	Water Temp. Inlet	Outlet	Vacuum	Barom	Remarks	
A.M.	12:10	175	15	2		60	11	95	118	60	80	28.6	27.92		Jol
	1:15	176	14½	2		57	10	95	117	57	79	28.6			Jol
	2:10	175	15	2		58	10	95	117	59	79	28.8			Jol
	3:10	180	15½	2		57	10	95	117	58	79	28.7			Jol
	4:05	178	15	2		59	10	95	117	59	79	28.7	27.88		Jol
	5:05	178	15	2		58	10	95	117	59	79	28.7			Jol
	6:00	176	15	2		58	10	95	117	59	79	28.6			Jol
	7:15	175	15	2		58	10	95	117	59	79	28.6			Jones
	8:15	176	15	2		58	10	95	117	59	79	28.7	29.86		Jones
	9:15	179	15½	2		58	10	96	118	60	80	28.6			Jones
	10:10	180	15½	2		57	10	96	118	60	80	28.6			Jones
	11:10	180	15	1½		58	10	97	120	61	81	28.5		No stand setting out	Jones
P.M.	12:10	179	15	1½		58	10	98	121	62	83	28.5	29.92		Jones
	1:05	179	15	1½		57	10	99	122	63	85	28.5			Jones
	2:10	178	15	1½		58	10	99	122	63	85	28.5		Watch H.P. stand.	Jones
	3:05	180	15	1½		57	10	99	122	63	85	28.5			Jones

FIG. 235.—Portion of a turbine-room log sheet upon which entries should be made at inspection time.

for unusual conditions and for taking instrument readings. Unusual operating conditions are generally evidenced by the oil temperatures, vacuum readings or by vibration or noise. Hence, by recording the various instrument readings hourly upon a log sheet (Fig. 235), troubles will generally become apparent as soon as they arise. If the bearing oil or governor-oil pressure should decrease materially, the auxiliary oil pumps should be started immediately. Whenever there is evidence of water in the turbine casing, open the drain valves to allow the water to escape. On turbines which have individual nozzle-control valves, the operator should always see to it that only the minimum number of nozzles required to carry the load is open.

As a rule, it will be found that the condenser and its auxiliaries will require much more attention than will the turbine itself.

Note.—Keeping Up The Vacuum On A Condensing Turbine is one of the most difficult tasks in connection with the operation of the turbine. A decreased vacuum is generally due to one of two causes: (1) *Air leaks.* (2) *Fouled tubes or nozzles.* To determine which of these factors has been the cause in any particular case is usually quite difficult. A scheme which is sometimes employed for finding the cause is to arrange some simple means for measuring the quantity of air discharged by the air pump. For this purpose either a gas meter may be used or a *pitot tube* (Fig. 236) may be inserted into the discharge pipe and connected to a draft gage. An increase in the quantity of air discharged will indicate new air leaks.

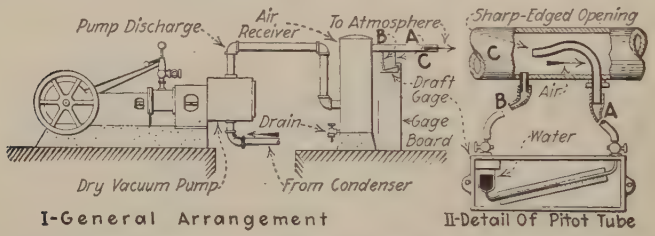


Fig. 236.—General arrangement and detail of pitot tube for measuring the quantity of air discharged by a dry-vacuum pump. The difference between the pressures in pipes *A* and *B* is a measure of the velocity of flow through the pipe *C* and hence indicates the volume of air discharged. An increase in the pressure difference, as read by the draft gage, indicates air leaks.

217. Shifting Loads From One Turbine To Another, when they are operating in parallel, is generally effected by varying the governor-spring tension. If it is desired to cause a certain turbine to take more load, the governor spring is adjusted as for greater speed; (see Div. 6). This will cause it to take a greater fraction of the total load. Likewise, to cause a certain turbine to take less loads, its governor is adjusted just as it would be for lesser speed. In electric-power plants, the division of the load is generally effected by the switchboard operator by his operation, from the switchboard, of the motor-operated governor-spring adjusting device (Sec. 151).

Note.—Working Its Field Rheostat Does Not Change The Power Load On An Alternator Which Is Operating In Parallel

with another alternator; it merely changes the valve of the cross-current between the two machines. To adjust for minimum cross-current, adjust the field rheostats so that the sum of the line-current-ammeter readings for the two machines will be a minimum.

218. To Stop A Turbine which is operating under load, it is customary to gradually decrease the load on the turbine, before shutting off the steam supply. This procedure is not essential, however, as no harm will result to the turbine if the steam supply is shut off while the machine is under load—harm may, however, result under some conditions to the machine which the turbine drives. The following procedure in stopping a turbine will apply in nearly all cases.

Procedure.—1. Start the auxiliary oil pump.

2. Gradually decrease the load on the turbine by varying the governor-spring tension; (Sec. 151). When the load is reduced to about one-tenth of full load, reduce the vacuum to 24–26 in. by opening the vacuum breaker valve. Remove the entire load if possible.

3. Pull the trip lever to close the emergency-governor valve and allow the turbine rotor to come to rest. See that the bearings are receiving oil while the rotor is stopping.

4. Stop the auxiliary oil pump.

5. After about 15 minutes stop the condenser pumps. This will insure that all water vapor is drawn from the turbine casing.

6. Open all drains and leave them open until the turbine is started again.

7. Close the turbine stop valve, X (Fig. 234), and open a drain between it and the throttle valve, T. This will prevent steam from blowing past the throttle valve and tending to cause leakage.

219. To Apply The Electrical Or Mechanical Load When Starting A Turbine, the following instructions will be found of value. It is assumed that the turbine which drives the load has been started as outlined in Secs. 214 and 215.

1. To Start A Single Alternator.—(*a*) Start the exciter and adjust for normal voltage. (*b*) Turn the generator field rheostat so that all of its resistance is in the field circuit. Close the field switch. (*c*) Adjust the rheostat of the exciter for normal voltage. Slowly increase the voltage to normal by cutting out the resistance of the field rheostat. (*d*) Close the main switch.

2. To Start An Alternator To Run In Parallel With Others.—(*a*) Adjust the exciter voltage and close the field switch, the resistance

being all in, as described above. (*b*) Adjust the generator field resistance so that the generator voltage is the same as the bus-bar voltage. (*c*) Synchronize the generator with the bus-bars—see the author's American Electrician's Handbook. Close the main switch. (*d*) Adjust the field rheostat until cross currents are a minimum and adjust the governor of the turbine so that the load will be distributed, as desired, among the operating generators.

3. To Start A Direct-current Generator.—(*a*) Before starting the turbine close the field switch and see that the entire rheostat resistance is in the field circuit. Then bring the machine up to speed. (*b*) Cut out field resistance to raise the voltage to the rated value or, if parallel operation is desired, to the voltage of the bus-bars. (*c*) Close the main line switch. (*d*) Adjust the load on the generator by varying its field resistance.

4. To Start A Centrifugal Pump.—(*a*) Before starting the turbine prime the pump and close its discharge valve. This will permit starting under fractional load. Then bring the machine up to speed. (*b*) Open the discharge valve gradually to put load on the pump. See also the author's Steam Power Plant Auxiliaries And Accessories.

220. To Take The Load Off Of A Turbine in stopping it, the procedure is generally the reverse of that which is performed in starting up and applying the load. To avoid misunderstanding the following instructions are given:

1. To Stop A Single Alternator.—(*a*) Decrease the field current by turning in all of the field-rheostat resistance. (*b*) Stop the turbine, (Sec. 218). (*c*) Open all switches and stop the exciter.

2. To Cut Out An Alternator Which Is Running In Parallel With Others.—(*a*) Partly close the turbine throttle valve so that the load on the generator is reduced. (*b*) Open the main switch. Do *not* open the field switch before opening the main switch. (*c*) Stop the turbine. (*d*) Open the field switch and stop the exciter.

3. To Stop A Single Direct-current Generator.—(*a*) See that all motors are disconnected from the lines. (*b*) Stop the turbine, (Sec. 218). (*c*) Turn all rheostat resistance into the field circuit. (*d*) Open the main switch.

4. To Stop A Direct-current Generator Operating In Parallel With Others.—(*a*) Reduce the load as much as possible by throwing all resistance into the field circuit with the field rheostat. (*b*) Throw off the load by opening the circuit-breaker, if one is used; otherwise open the main switch. (*c*) Stop the turbine, (Sec. 218).

5. To Stop A Centrifugal Pump,—(*a*) If the pump is operating in parallel with others, close the discharge valve. (*b*) Stop the turbine, (Sec. 218).

221. Regular Inspections Of Steam Turbines Should Be Made.—The object of such inspections is to find the source of some possible trouble before the trouble actually shows itself. Since all turbines operate at high speeds and with only rotating motions, slight amounts of wear will not give warning as by knocks or the like, but will increase until some serious damage occurs—such as the rubbing of blades or the burning out of a bearing. To forestall such damage, the following inspections are recommended.

1. HOURLY INSPECTIONS.—Hourly readings should be taken of the temperatures and pressures of the oil at various points in the system, the temperatures of the circulating water and condensate, the vacuum in the condenser, the pressure and superheat of the supply steam, steam pressures in various stages of the turbine, load on the turbine, and other like quantities. These readings, together with any unusual noise or circumstances, should be recorded on a log sheet (Fig. 235) which is kept for the purpose. Irregularities in any of these readings will immediately disclose some approaching trouble.

2. MONTHLY INSPECTIONS.—At least as often as once a month, a test should be made on the emergency governor by gradually increasing the speed of the turbine above normal rated speed to that at which the governor should shut off the steam supply. If the governor operates, the speed should be recorded. If the governor does not operate it should be adjusted or repaired. The steam strainer should be inspected, cleaned if necessary, or if in poor condition it should be replaced. The alignment of the unit should be checked very carefully. In some installations, measurements are made each month for possible settling of the foundation. The adjustment of the thrust bearing should also be checked, (Div. 5).

3. YEARLY INSPECTIONS.—Once each year the entire unit should be dismantled, cleaned, and all parts inspected for wear. The steam passages should be carefully examined for erosion. Badly worn valves, nozzles, or blades should be replaced if possible. It is to be expected that, after a number of years of service, the parts which are subjected to the action of steam flow will be worn quite badly. In such cases, new parts should be obtained from the manufacturer. When the parts are again assembled, all bearings should be adjusted (see Div. 5) so as to obtain the proper clearances and alignment.

222. The Maintenance Of Steam Turbines, aside from the periodic inspections, Sec. 221, involves only: (1) *Keeping up the purity and quality of the oil;* this is treated fully in Div. 10. (2) *Making adjustments and replacements;* the bearings should

always be so adjusted that the alignment and clearances are correct; worn bearings, which will scarcely ever be found if the lubrication and alignment are carefully attended to, may be rebabbitted (Sec. 97) or replaced; badly worn nozzles, blades, or valves should be replaced.

Note.—The Repair Of Broken Blading should not be attempted by the turbine operator. Such repairs should be made by the manufacturer of the turbine, because it is essential that the repaired blading be tested for strength and balanced before being put to service. Sometimes, if a unit on which some blades have broken cannot be spared from service for some time, a temporary repair can be effected by cutting out all of the blades which remain in the rows from which some have been lost. This will restore the balance of the rotor and will permit running the turbine at a slightly reduced capacity and with but a slight loss of efficiency. Later, when the unit can be spared and the manufacturer is ready to make the repairs, the rotor may be shipped to his factory.

223. If A Turbine Will Not Carry The Load Which It Should, the cause is most probable one of the following (*Terry Instruction Book*):

1. Excessive Load.—(*a*) Overloaded driven machine. (*b*) More power required than the turbine was built to develop. (*c*) Wear of driven machine has lowered efficiency, requiring more power.

2. Plant Conditions.—(*a*) Steam pressure at the throttle less than that stamped on the nameplate. (*b*) Turbine designed for superheat but run on saturated steam. (*c*) Turbine designed for dry steam, but very wet steam used. (*d*) Back pressure in casing greater than specified. (*e*) On condensing turbine, vacuum is low.

3. Turbine Adjustments.—(*a*) Hand valves closed that should be open. (*b*) Governor closes valve before normal speed is reached. (*c*) Valve improperly set, (see Div. 6). (*d*) One or more jets plugged. (*e*) Clogged strainer in steam line. (*f*) "Lap" or "Clearance" wrong, (see Div. 5). (*g*) Buckets worn by wet steam or otherwise. If so, describe conditions to manufacturers and they will advise. (*h*) Parts binding or rubbing. (*i*) If turbine has been taken apart the wheel may be on backwards or, in a multi-stage turbine diaphragms or wheels interchanged.

224. If The Steam Consumption Of A Turbine Becomes High, the probably causes (*Terry Instruction Book*) are:

1. The Same Causes As For Insufficient Power, (Sec. 223).

2. Hand Valve Control.—(*a*) Keep as many hand valves closed as load conditions will allow, and thus keep the pressure in the steam ring as high as possible, to get the best use of the steam pressure available.

(*b*) Do not run with hand valves "cracked." Keep them either open or shut. (*c*) Inspect hand valve seats. Leakage here will cause loss when valves are closed.

3. IF THE TURBINE RUNS BELOW SPEED, the water rate will be increased and the capacity decreased. In the case of pumps running from a pressure governor, however, the overall efficiency of the unit is benefited by running at reduced speed when lightly loaded, on account of reduced pump losses.

225. When Writing To The Factory For Advice, the following information should be given (*Terry Instruction Book*):

1. When writing to the Terry Steam Turbine Company, in regard to the power or economy of a turbine, please read the above tabulation (Secs. 223 and 224) and so far as possible advise us on the various points covered.

2. Take readings as follows with the turbine running under load, repeating for several loads if possible: (*a*) Steam pressure in the steam line between the throttle valve and the turbine. (*b*) Superheat or moisture in the steam. (*c*) Steam pressure in the steam ring. (*d*) Number and position of hand valves open and closed. (*e*) R.p.m. of turbine. (*f*) Back pressure at the turbine exhaust. (*g*) Load on driven machine, if measurable.

3. Give all information on the name plate of the machine especially the serial number.

QUESTIONS ON DIVISION 11

1. State the three fundamentals of steam-turbine operation in the order of their importance.

2. What precautions should be observed if successful operation of a turbine is to be attained?

3. List about 5 points which affect the safety of a turbine's operation.

4. What factors tend for uninterrupted service in turbine operation?

5. How should a turbine be operated to insure maximum economy?

6. State the five principal troubles which are likely to arise in the operation of a turbine. How are they guarded against?

7. What are the eight principal causes of turbine vibrations? How would you distinguish which is the cause in any particular case?

8. What special precautions should be exercised in starting a newly installed turbine for the first time.

9. Give the steps required in starting a non-condensing turbine. Illustrate with a sketch.

10. Should the turbine rotor be turning when the steam is turned on to warm it? Why?

11. State the procedure of starting a condensing turbine. Illustrate with a sketch. Is the turbine started under a vacuum or non-condensing?

12. What special procedure should be followed in starting a bleeder or a mixed-pressure turbine?

13. What care does a turbine require while it is running?

14. What are the two principal causes of a gradual decrease in the vacuum on a turbine? How may an operator distinguish the actual cause in a given case?

15. Make a sketch of and describe the pitot-tube method of measuring air discharge.

16. Should a turbine be stopped with the load on or after removing the load? Why?

17. State the usual procedure of stopping a turbine.

18. Describe the methods of applying the load to a turbine with (*a*) a single alternator, (*b*) an alternator which is to run in parallel with others, (*c*) a direct-current generator, (*d*) a centrifugal pump.

19. Explain how to take off the load from a turbine which is driving (*a*) a single alternator, (*b*) an alternator which is running in parallel with others, (*c*) a single direct-current generator, (*d*) a direct-current generator in parallel operation with others, (*e*) a centrifugal pump.

20. What are the purposes of making regular inspections of a turbine?

21. Explain what should be done at each *hourly* inspection.

22. Explain what should be done at each *monthly* inspection.

23. Explain what should be done at each *yearly* inspection.

24. What are the essential points in the maintenance of a steam turbine?

DIVISION 12

STEAM-TURBINE TESTING

226. The Purpose Of Testing A Steam Turbine For Performance is to obtain data whereby the performance values, or heat economy, may be computed (Sec. 240). The performance values which are computed from the results of the test may be used in determining: (1) *How nearly the performance of the turbine approaches or exceeds that which was guaranteed by the manufacturer.* A test for this purpose is called an acceptance test. (2) *Whether or not an old turbine is operating at its maximum efficiency.* (3) *The comparative performance of two or more prime movers.* (4) *The overall economy of the power plant.* Various methods of testing steam turbines are described hereinafter in this division.

NOTE.—THE CONDITIONS UNDER WHICH A TEST IS MADE SHOULD BE GOVERNED BY THE OBJECT OF THE TEST. Turbines are usually sold under a guarantee (Sec. 285) which is based upon certain operating conditions, such as the initial and final conditions of the steam, speed of rotation, and load. Consequently, if the results of a test are to be used in comparing the actual operating performance with the guaranteed performance, the conditions under which the test is made should conform as nearly as possible to those specified in the guarantee. However, if the object of the test is to compare the performances of two prime movers on an economic basis, the test of each should be made under the conditions for which it was designed. Then, a correction (Sec. 268) should be made to reduce both performances to the same, comparable, basis. In testing a turbine to determine the overall economy of a power plant, the conditions under which the test is made should, as nearly as is possible, conform to the conditions under which the plant normally operates.

227. The More Important Data Obtained In Testing A Steam Turbine are: (1) *Condition of the steam entering the turbine.* (2) *Condition of the steam at the turbine exhaust.* (3) *Power output of the turbine.* (4) *The quantity of steam*

consumed by the turbine. (5) *The speed of rotation of the turbine shaft.* Various methods of obtaining these data are described in following sections of this division.

228. The Duration Of A Steam-turbine Test should ordinarily be from 3 to 5 hr. However, the object of the test (Sec. 226) may render it desirable to extend the test period over a longer time. A test over a period of less than about 3 hr. cannot be relied upon for accurate results. The readings of the various instruments should be made and recorded (Fig. 245) at intervals of not more than 30 min.

Note.—The Duration Of A Steam-turbine Test As Specified By The A.S.M.E. Code is quoted below. Where practicable, this specification should be followed: "A test for steam or heat consumption, with substantially constant load, should be continued for such time as may be necessary to obtain a number of successive hourly records, during which the results are reasonably uniform. For a test involving the measurement of feed water for this purpose, 5-hr. duration is sufficient. Where a surface condenser is used, and the measurement is that of the water discharged by the condensate pump, the duration may be somewhat shorter. In this case, successive half-hourly records may be compared and the time correspondingly reduced. When the load varies widely at different times of the day, the duration should be such as to cover the entire period of variation."

229. The Apparatus And Instruments Which Are Required For Testing A Steam Turbine depend upon the object of the test (Sec. 226), and upon the local conditions and arrangement of the plant. In general, however, those instruments which are listed in the A.S.M.E. Code (Sec. 248) should be available. All instruments which are used should be accurately calibrated according to the rules of the A.S.M.E. Code before and after each test. Then, the observations should be corrected for any errors which may be noted in the instrument readings.

230. The Condition Of The Steam Entering The Turbine Is Determined by: (1) *The pressure*, in pounds per square inch, which is read from a pressure gage, P, Fig. 237. (2) *The temperature of superheat or the quality.* The temperature of the steam is determined by a thermometer (Fig. 238 and T, Fig. 237). Then, from a steam table, determine the temperature of saturated steam at the pressure indicated by

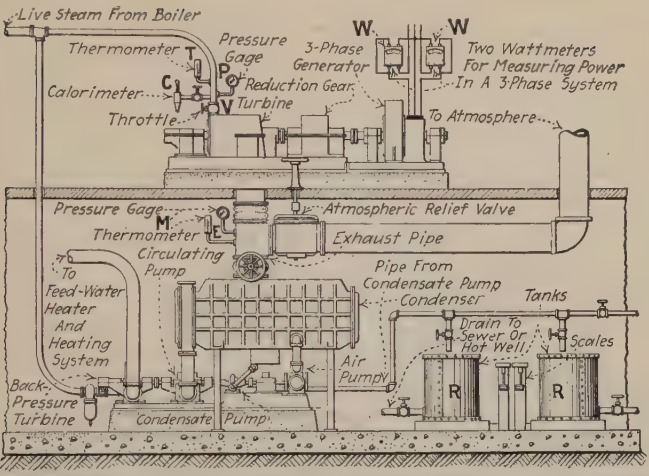


FIG. 237.—Illustrating arrangement of apparatus for testing a small-capacity steam turbine driving a three-phase generator and exhausting into a surface condenser.

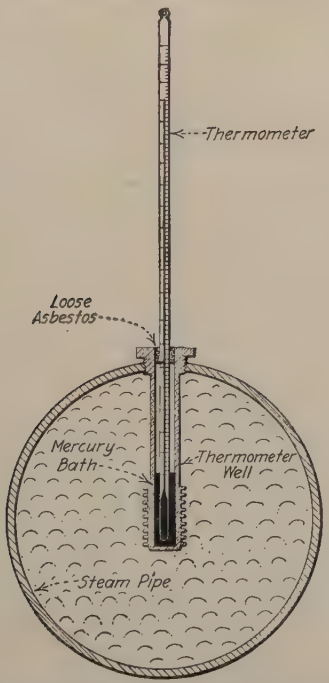


FIG. 238.—Showing method of obtaining the temperature of steam which is flowing in a pipe. (The length of the well should be such that the bulb of the thermometer will be at about the center of the pipe.) Unless the thermometer which is used is one that is graduated for the specific "immersion," its readings should be corrected for "stem exposure;" see the author's PRACTICAL HEAT.

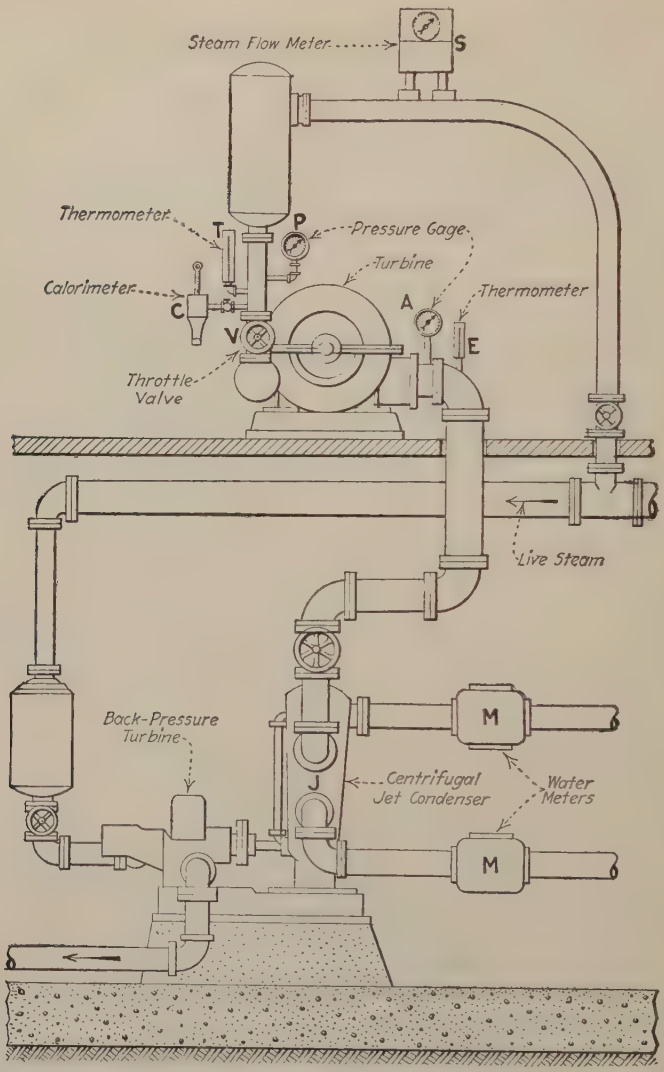


Fig. 239.—Illustrating location of apparatus for testing a steam turbine which has a back-pressure-turbine-driven jet condenser, *J*. The steam consumption is determined by a steam flow meter, *S*, or by water meters, *M*.

the pressure gage, P. If the temperature as read from the thermometer (T, Fig. 237) is more than about 10° F. above that found in the steam table corresponding to the reading of P, it is reasonably certain that the steam is superheated. The temperature of the superheat will then be the difference between the temperature as read on the thermometer and the temperature of the steam as obtained from the steam table. If the difference between the thermometer reading and the temperature of saturated steam as obtained from the steam table is less than 10 °F., the steam may be wet, and its quality should be determined by a calorimeter, C, Fig. 237.

Note.—The Location Of The Instruments For Determining The Condition Of The Steam Entering The Turbine should be as near to the steam-inlet flange as is practicable (see Figs. 237 and 239). The throttle, V, Fig. 237, should be wide open during the test.

231. The Property Of The Steam At The Turbine Exhaust Which Must Be Determined Is The Temperature. However, as stated below, both the temperature and the pressure of the exhaust steam are usually noted. The temperature of the steam is determined by inserting a thermometer (Fig. 238) in the exhaust pipe of the turbine. This thermometer (E, Figs. 237 and 239) should be located as near as is practicable to the turbine exhaust flange.

Note.—If The Exhaust Pressure Or The Condenser Pressure Is Determined By A Pressure Gage—of either the Boudon-tube or mercury-column type—the reading of this gage should be recorded as referred to a barometric pressure of 30 in. of mercury. That is, if during the tset the barometric pressure is 29.5 in. of mercury, and the pressure gage indicates a condenser pressure of 27.5 in. of mercury, *the condenser pressure referred to a 30-in. barometer* = 30 − (29.5 − 27.5) = 30 − 2 = 28 *in. of mercury.* Thus, the condenser pressure as referred to a 30-in. barometer results in the pressure which would be indicated by the vacuum gage if the atmospheric pressure were 30 in. of mercury. For accurate results, a mercury-column pressure gage should be used for determining the exhaust and condenser pressures. The barometric pressure should be determined by a barometer which is located near the pressure gage. If no barometer is available for reference, the barometric reading may be obtained from the local Weather Bureau.

232. The Power Output Of The Turbine May Be Determined: (1) *Mechanically*, by a brake, such as a prony brake, or

a water brake. For methods of obtaining the power output by a brake see the author's Steam-engine Principles and Practice. (2) *Electrically*, by measuring the electrical energy or the power output of the driven generator. This method, which is described in the following sections, is practically always employed in testing turbo-generators.

233. The Power Output Of A Turbo-generator May Be Determined Electrically At The Generator Terminals by wattmeters, ammeters and voltmeters, or watt-hour meters. Whichever instruments are used should be of the portable type, and should be so screened that they will not be affected by any stray magnetic fields. If the load remains practically constant throughout the test, the use of wattmeters (*W*, Fig. 237) will generally result in greater accuracy than will the use of a watt-hour meter. However, if during the test, the load fluctuates materially, a watt-hour meter should be used. Then, the average power output, in kilowatts, may be determined by dividing the number of kilowatt-hours, as indicated by the watt-hour meter, by the number of hours duration of the test.

That is:

$$(31) \qquad Av.\ kw.\ power\ output = \frac{Kw.\text{-}hr.\ generated\ during\ test}{Hours\ duration\ of\ test} \quad \text{(kilowatts)}$$

Example.—If during a certain test, of 4-hr. duration, 4,876 kw.-hr. of energy are generated, what is the average power developed during the test? Solution.—Substitute in For. (31): *Av. kw. power output* = (*Kw.-hr. generated during test*)/(*Hours duration of test*) = 4,876 ÷ 4 = 1,219 *kw.*

Note.—In Measuring The Power Output Of A Turbo-alternator Electrically, it is preferable that the load on the alternator be as near unity power factor as is possible. The reason for this is, that if the power factor of the load is unity, the error which would otherwise be caused by phase displacement in the instrument transformers will be obviated. A load at practically unity power factor may be obtained by connecting the generator to a water rheostat. If a three-phase alternator is operating under an inductive load—power factor less than unity—the proper balancing of the load on each of the three phases should be checked by the station ammeters and voltmeters. For detailed instructions for measuring the electrical output of generators, see the author's Steam-engine Principles and Practice.

234. The Power Output Of A Generator Should Be Determined As The "Net Watts" Output.—That is, the power required for excitation should be recognized in determining the power-output value of the generator. Thus, if the generator is self-excited (direct-current generator) or if the exciter is direct-connected to the turbo-generator shaft (as it is on some turbo-alternators) the energy for excitation need not necessarily be considered. However, if the generator is separately excited, the power, in watts, which is supplied to the generator for excitation must be measured. Then, to obtain the net power output of the generator subtract the power input required for excitation from the power output as measured at the generator terminals, see colums 5, 6 and 7 (Fig. 245). That is, for a separately-excited generator:

(32) *Net kw. output* = (*Kw. output at terminals*) − (*Kw. excitation*) (kilowatts)

235. The Quantity Of Steam Consumed By The Turbine Is Generally Determined By One Of The Following Methods: (1) *By measuring the condensate.* (2) *By measuring the feed water.* (3) *By a steam-flow meter.* The first method—that of weighing the condensate—will, generally, result in greater accuracy than will any of the other methods. Consequently, where practicable, it should be used. Each method is described in a following section.

236. To Determine The Quantity Of Steam Consumed By A Turbine By Measuring The Condensate Discharged From A Surface Condenser, the water which is discharged by the condensate pump is generally piped (Fig. 237) to tanks, *R*, which set on weighing scale platforms and is there weighed. Thus, by weighing the condensed steam which is discharged during a certain number of hours, and dividing the total weight by the number of hours, the total steam consumption, in pounds per hour, results. That is:

$$(33) \qquad \textit{Total lb. per hr. steam consumption} = \frac{\textit{Lb. condensate discharged during test}}{\textit{Hours duration of test}} \quad \text{(pounds)}$$

This method of determining the steam consumption is only

practicable where the turbine is operated (Fig. 237) in conjunction with a *surface* condenser. The arrangement of tanks,

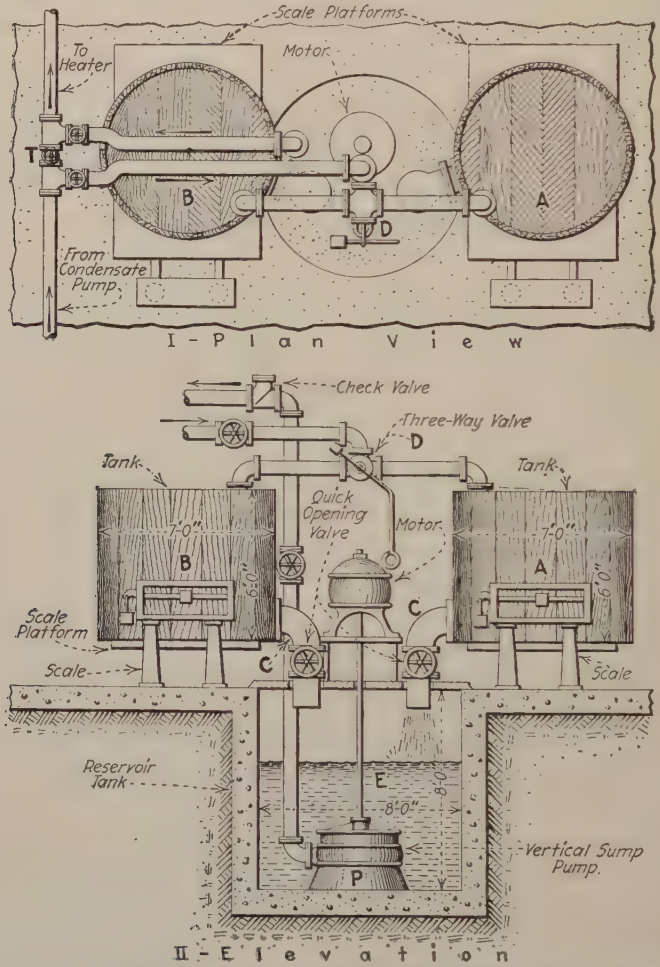


Fig. 240.—A convenient arrangement of tanks and piping for weighing the condensate from a surface condenser.

piping, and scales for weighing the water, and corrections which must be made are treated in the notes below.

Note.—A Tank And Piping Layout For Weighing The Condensed Steam Which Is Discharged From The Surface Condenser

is shown in Fig. 240. The discharge pipe from the condensate pump is tapped at T (Fig. 240-*I*) and an arrangement is made as shown for by-passing the water through the tanks A and B. After one tank, A, has filled, the water from the condensate pump may, by means of the quick-opening three-way valve, D, be diverted into the other tank, B. Then while B is filling, the water in A may be weighed. After it has been weighed the quick-opening valve in the large outlet pipe, C, is opened, so that by the time B is full, all of the water that was in A has been weighed and discharged into the reservoir tank, E. The water is removed from E by the vertical motor-driven centrifugal pump, P. The dimensions of the tanks as shown in Fig. 240-*II* should provide sufficient capacity for testing a 5,000-kva. turbine. In the event that scales are not available for weighing the water which is discharged from the condenser, its weight may be computed by the following formula:

$$W = AhD \quad \text{(pounds)} \tag{34}$$

Wherein: W = the weight, in pounds, of the water in the tank. A = the cross-sectional area of the tank, in square feet. h = depth, in feet, of the water in the tank. D = the density of the water, in pounds per cubic foot, at the temperature of the water in the tank. To obtain D, it is necessary to measure the temperature of the water in the tank. Then from a table of densities of water (this is given in most steam tables), find the density in pounds per cubic foot at the measured temperature.

Note.—In Measuring The Condensate From A Surface Condenser, The Amount Of Leakage Of Either From The Condenser-circulating-water Passages Or From Other Sources (Sec. 248) Must Be Determined And Proper Corrections Made.—One method of determining the condenser leakage is to raise the vacuum in the condenser to the operating value and, with the throttle (V, Fig. 237) closed, determine the amount of water which is discharged by the condensate pump. This test of condenser leakage should be continued for a period of at least 2 hr. If the leakage test results in an appreciable amount of water being discharged from the condensate pump, the leaks in the condenser should be located and repaired before proceeding with the turbine test. This is because that, when the turbine is operated at full load, the leakage will be much greater than it was when the leakage test was made with the throttle closed. There are other methods of determining the condenser leakage at full load, such as by chemical titration or by electrical resistance, but they will not be described herein. Any water leakage into or out of the condenser from the turbine or pump glands must be determined and proper correction made therefor.

237. To Determine The Quantity Of Steam Consumed By A Turbine By Measuring The Boiler Feed Water (Fig. 241), the water may be piped from the feed-water heater, H, to

tanks *A* and *B*, which are supported on weighing-scale platforms, where it is weighed. The water is, after weighing, emptied into the reservoir, *R*. From *R* it is pumped to the boiler by the boiler feed pump, *P*. The water level in the boiler, as indicated by the water gage thereon, should be the same at the end of the test as it was at the beginning. Then, by deducting the leakage (see note below) from the total weight of the water pumped into the boiler during the test, the steam consumption for the duration of the test results.

Note.—In Determining The Steam Consumption By Measuring The Boiler Feed Water, The Leakages For Which Corrections

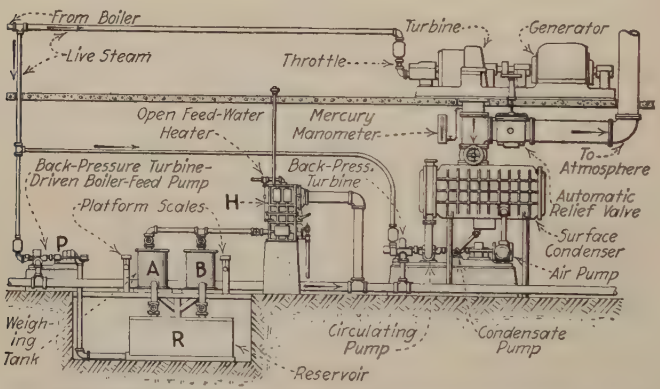


Fig. 241.—Showing arrangement of tanks for weighing boiler feed water.

Must Usually Be Made Are: (1) *The leakage of water which occurs in the boiler feed pump, and in the pipes between the reservoir* (*R*, Fig. 241) *and the boiler.* The amount of this leakage may be determined by closing off all the feed valves at the boiler, running the pump, *P*, for about 15 min., and noting the quantity of water which has disappeared from the supply tank, *R*. In making this test, a pressure gage should be placed in the pump discharge to guard against a dangerous water pressure in the pipe. During this leakage test, the reading of this pressure gage should be approximately that of normal operation when the feed valves are open. (2) *The leakage of steam from the boiler, and from the connections and valves between the boiler and the turbine.* This leakage may be determined by shutting off the feed-water supply, and by breaking and blanking off all branch connections to the steam line which connects the boiler to the turbine. Then, by means of a slow fire, maintain the steam pressure at the same pressure which is to obtain during the test. This pressure

should be maintained for a period of at least 2 hr., and the water level in the gage glass should be noted at about 15-min. intervals. The amount of steam which has leaked out may be computed by the amount of the decrease in the water level as shown by the water gage. For more detailed instructions concerning these leakage tests, see the A. S. M. E. Test Code (Sec. 248).

238. A Steam-flow Meter May Be Used To Determine The Steam Consumption Of A Turbine.—The meter should be connected into the high-pressure steam line (*S*, Fig. 239) near the turbine. It should be calibrated in that place with approximately the same temperature, pressure and steam flow as will obtain during the subsequent turbine test. A steam-flow meter cannot generally be depended upon for an accuracy of more than about 2 per cent. Where accurate test results are desired it should not be used.

Note.—Water Meters May Be Used To Measure The Water For Determining Steam Consumption, either in the boiler feed-water line or in the condensate-pump discharge. The condensate in a jet condenser, *J*, may be determined (Fig. 239) by metering the injection water and the discharge water, and then taking their difference. Water meters, *M*, when used, should be frequently calibrated in place. They cannot be depended upon for accurate results.

239. The Speed Of Rotation Of A Turbine Rotor Is Generally Determined By A Tachometer (Figs. 242 and 243). If the power output is to be determined electrically (Sec. 233), the only purpose of the tachometer is to insure that the rated speed is maintained constant throughout the test. For a turbo-alternator which is equipped with a frequency meter, the tachometer may be dispensed with. However, if the power output is to be determined by the brake method (Sec. 232) an accurately-calibrated tachometer is essential.

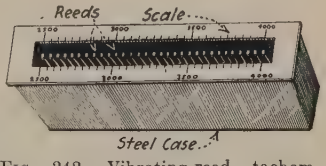


Fig. 242.—Vibrating-reed tachometer. (*Jas. G. Biddle Co.*)

240. The Various Terms And Efficiencies Which Are Generally Used To Express Steam-turbine Performance Values (Sec. 226) are discussed and explained in following

Secs. 241 to 245. The terms are: (1) *The water rate*, which is expressed as the number of pounds of steam required to generate a kilowatt-hour or a horsepower-hour of energy. A water rate graph is shown in Fig. 244. If the turbine is used to drive a generator, the water rate is usually expressed in pounds of steam per kilowatt-hour. If used to drive a pump, compressor, or the like, the water rate is usually expressed in pounds of steam per brake horsepower-hour. Whenever the water rate of a turbine is given as its performance value, the

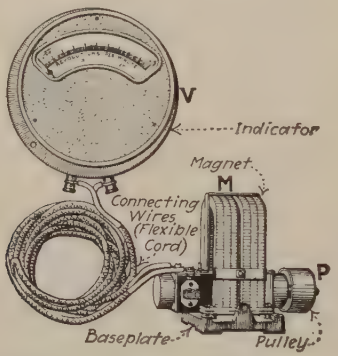


Fig. 243.—Electric tachometer. (The tachometer consists chiefly of a direct-current magneto, *M*, and a voltmeter, *V*. The pulley, *P*, is driven by belt from the shaft the speed of which is to be measured. Since the magnetic field of *M* is produced by permanent magnets, the voltage which it generates will be proportional to its speed. Hence, the scale of *V* can be calibrated to indicate revolutions per minute directly.)

steam conditions at inlet and exhaust should also be given; unless the steam conditions, are stated, the water rate is a very indefinite performance value. (2) *The number of heat units required to develop one unit of mechanical or electrical energy*, which is expressed as the number of British thermal units per kilowatt-hour or per brake horsepower-hour. (3) *The net mechanical work developed by one heat unit*, which is expressed in foot-pounds of net work per British thermal unit. (4) *The thermal efficiency*, expressed as a percentage. (5) *The Rankine cycle ratio* expressed as a percentage. The example given below is merely to illustrate the method of computing the above performance values from assumed test data of a turbo-alternator, and is not intended to represent the performance of any particular machine.

Example.—The half-hourly observations of a full-load test on a 10,000-kva. turbo-alternator are as recorded in Fig. 245. Compute the following performance values: (*a*) *The water rate.* (*b*) *The number of British thermal units consumed per kilowatt-hour.* (*c*) *The number of foot-*

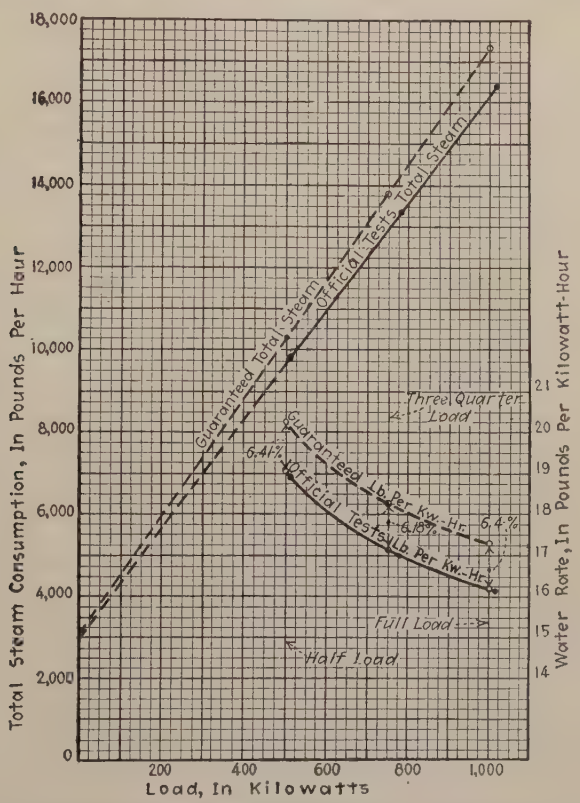


Fig. 244.—Graph showing total steam consumption and the water rate of a 1,000-kw. steam turbine. (Dotted lines show the guaranteed consumption. Full lines show consumption, as determined by official test. The graphs are for a 1,000-kw., 3,600-r.p.m., turbine, for the City of Grand Rapids, Michigan, operating under the following steam conditions: Dry saturated steam at a pressure of 140 lb. per sq. in., gage, and a vacuum of 28 in. of mercury, referred to a 30-in. barometric pressure. *Allis-Chalmers Mfg. Co.*)

pounds of net work developed per British thermal unit. (*d*) *The thermal efficiency.* (*e*) *The Rankine cycle ratio.*

Solution.—The averages of the half-hourly data readings, as recorded, are computed and entered in the last line of Fig. 245. The values of $\mathbf{H}_1$ and $\mathbf{H}_2$ (at bottom of Fig. 245) are 1,252 and 894 B.t.u. respectively, determined from a steam chart (Fig. 15) on the basis of the supply steam

at a pressure of 150.8 lb. per sq. in. gage and a superheat (Sec. 230) of 100.4° F. at the throttle, and the exhaust steam at a temperature of 92.8° F. The value of $\mathbf{h}_2$, 61 B.t.u., as determined from a steam table, is the number of British thermal units in 1 lb. of water at the temperature (92.8° F.) of the exhaust.

(*a*) The Water Rate May Be Determined by the following formula:

$$\mathbf{W}_K = \frac{\mathbf{W}}{t\mathbf{P}_K} \qquad \text{(lb. per kw.-hr.)} \tag{35}$$

Wherein: $\mathbf{W}_K$ = the weight of steam, in pounds, required to develop 1 kw.-hr. $\mathbf{W}$ = the total weight of steam, in pounds, consumed by the turbine for the duration of the test. t = the duration of test, in hours. $\mathbf{P}_K$ = average net power, in kilowatts, developed by the turbine. From Column 9, Fig. 245, $\mathbf{W}$ = 436,800 *lb.* From Column 1, Fig. 245, t = 3

1 TIME	LIVE STEAM 2 Pressure, Lb. Per Sq. In. Gage	3 Superheat, °F. Or Quality in %	4 TEMP OF EXH. STEAM °F	POWER OUTPUT 5 Generator Kw.	6 Exciter Kw.	7 Net Power Kw.	8 R.P.M.	9 TOTAL STEAM CONSUMPTION CORRECTED FOR LEAKAGE, LB.
2:00	153	102	92.5	10,200	300	9,900	4800	436,800
2:30	152	105	93.1	9,750	300	9650	4800	
3:00	152	98	92.8	10,100	300	9,800	4800	
3:30	148	101	93.0	10,100	300	9,800	4800	
4:00	149	99	92.8	10,150	300	9,850	4800	
4:30	150	98	92.6	10,200	300	9900	4800	
5:00	152	100	92.8	10,200	300	9,900	4800	
Average	150.8	100.4	92.8	10,128	300	9,828	4800	

The Following Is To Be Determined From Steam Chart Or Tables:
H_1 = Heat (Above 32° F) In 1 Lb. Of Steam At Throttle = 1252 B.T.U.
H_2 = Heat In 1 Lb. Of Steam At Exhaust (Assuming Adiabatic Expansion) = 894 B.T.U.
h_2 = Heat In 1 Lb. Of Water At Exhaust Temperature = 61 B.T.U.

Fig. 245.—Showing a form of log sheet for arrangement of steam-turbine test data for convenience in computing performance valves. (All of the heat values are above 32°F.)

hr. From Column 7, Fig. 245, $\mathbf{P}_K$ = 9,828 *kw.* Therefore, by For. (35), *the water rate,* $\mathbf{W}_K = \mathbf{W}/(t\mathbf{P}_K) = 436{,}800 \div (3 \times 9{,}828) = 14.8$ *lb. per kw.-hr.*

(*b*) The Number Of British Thermal Units Consumed Per Kilowatt-hour may be computed by the following formula:

$$\mathbf{Q} = \mathbf{W}_K(\mathbf{H}_1 - \mathbf{h}_2) \qquad \text{(B.t.u. per kw.-hr.)} \tag{36}$$

Wherein: $\mathbf{Q}$ = the heat, in British thermal units, consumed per kilowatt-hour. $\mathbf{W}_K$ = the water rate, in pounds of steam per kilowatt-hour. $\mathbf{H}_1$ = the heat, in British thermal units, in 1 lb. of steam at the throttle. $\mathbf{h}_2$ = heat, in British thermal units, in 1 lb. of water at the temperature of the exhaust. As stated in Fig. 245, $\mathbf{H}_1$ = 1,252 *B.t.u.*, and $\mathbf{h}_2$ = 61 *B.t.u.* From solution under For. (35), $\mathbf{W}_K$ = 14.8 *lb. per kw.-hr.* Therefore, by For. (36), *the heat consumed per kilowatt-hour* = $\mathbf{Q} = \mathbf{W}_K(\mathbf{H}_1 - \mathbf{h}_2) = 14.8 \times (1{,}252 - 61) = 17{,}626$ *B.t.u. per kw.-hr.*

(*c*) THE NUMBER OF FOOT-POUNDS OF NET WORK DEVELOPED PER BRITISH THERMAL UNIT may be computed by the following formula:

$$W = \frac{2{,}655{,}000}{Q} \qquad \text{(ft.-lb. per B.t.u.)} \tag{37}$$

Wherein: W = the work, in foot-pounds, developed by 1 B.t.u. Q = heat, in British thermal units, consumed per kilowatt-hour. 2,655,000 = the mechanical equivalent, in foot-pounds, of 1 kw.-hr. From the solution under For. (36), Q = 17,626 *B.t.u. per kw.-hr.* Therefore, by For. (37), *the number of foot-pounds of net work developed per British thermal unit,* W = 2,655,000/Q = 2,655,000 ÷ 17,626 = 150 *ft.-lb. per B.t.u.*

(*d*) THE THERMAL EFFICIENCY BASED ON NET GENERATOR OUTPUT (at generator terminals) may be computed by the following formula:

$$E_T = \frac{3{,}413}{Q} \qquad \text{(decimal)} \tag{38}$$

Wherein: E_T = the thermal efficiency, expressed decimally. Q = the heat, in British thermal units, consumed per kilowatt-hour. 3,413 = the heat equivalent, in British thermal units, of 1 kw.-hr. From solution under For. (36), Q = 17,626 *B.t.u. per kw.-hr.* Therefore, by For. (38), *the thermal efficiency,* E_T = 3,413/Q = 3,413 ÷ 17,626 = 0.193, or 19.3 *per cent.*

(*e*) THE RANKINE-CYCLE RATIO MAY BE DETERMINED by the following formula:

$$E_r = \frac{E_T(H_1 - h_2)}{(H_1 - H_2)} \qquad \text{(decimal)} \tag{39}$$

Wherein: E_r = the Rankine-cycle ratio, expressed decimally. E_T = thermal efficiency, expressed decimally. H_1, H_2, and h_2 are as specified in Fig. 245. From solution under For. (38), E_T = 0.193. From Fig. 245, H_1, H_2, and h_2 = 1,252, 894 and 61 *B.t.u.*, respectively. Therefore, by For. (39), *the Rankine cycle ratio,* $E_r = [E_T(H_1 - h_2)]/(H_1 - H_2) = [0.193(1{,}252 - 61)] \div (1{,}252 - 894) = 0.193 \times 1{,}191 \div 358 = 0.642$, or 64.2 *per cent.*

NOTE.—THE COMPUTATION OF THE PERFORMANCE VALUES OF A TURBINE ON THE BASIS OF THE BRAKE HORSEPOWER OUTPUT may be made in a manner which is substantially the same as that indicated in the solution of the above example. The brake horsepower is found by means of a brake (see the author's STEAM-ENGINE PRINCIPLES AND PRACTICE). Then, the value of the brake horsepower or its equivalent is used in the following formulas and For. (39).

$$W_H = \frac{W}{tP_H} \qquad \text{(lb. per hp.-hr.)} \tag{40}$$

$$Q_H = W_H(H_1 - h_2) \qquad \text{(B.t.u. per hp.-hr.)} \tag{41}$$

$$W = \frac{1{,}980{,}000}{Q_H} \qquad \text{(ft.-lb. per B.t.u.)} \tag{42}$$

$$E_T = \frac{2{,}545}{Q_H} \qquad \text{(decimal)} \tag{43}$$

Wherein: W_H = the weight of steam, in pounds, required to develop 1 hp.-hr. W = the total weight of steam, in pounds, consumed by the turbine during the test. t = the duration of the test, in hours. P_H = average net power, in horsepower, developed by the turbine. Q_H = the heat, in British thermal units, consumed per horsepower-hour. W = the work, in foot-pounds, developed by 1 B.t.u. E_T = the thermal efficiency, expressed decimally. H_1 and h_2 are as specified under Fig. 245.

241. The Reason Why The Five Different Methods Of Expressing The Performance Values Of Steam Turbines (Sec. 240) are used in the A.S.M.E. Test Code (Sec. 248) is that each method has a somewhat different significance. Each is discussed below. No one method has been adopted as a standard. Furthermore, various engineers prefer different bases for comparing the performance values of heat engines. Also an internal combustion engine does not have a water rate or a Rankine-cycle ratio. Hence, methods (2), (3) and (4) of Sec. 240 provide the only basis for comparing the thermal performance of a steam engine or a steam turbine with that of an internal combustion engine. Consequently, to provide for every contingency, a complete turbine test report should show each of the above mentioned (Sec. 240) performance values. See notes below and Secs. 242 to 245.

Note.—The Water Rate is generally used by turbine manufacturers as the basis of their performance guarantees. However, unless the initial and final steam conditions are known the water rate is meaningless. The reason is that the water rate for a given turbine will vary considerably with the steam conditions. It is used principally because all of the other performance values are determined from it; see Fors. (35) to (39). Furthermore, the average turbine purchaser has, through "handed-down" practice, learned to think of steam prime mover economies in terms of water rate. Where two turbines operate under the same steam conditions, their water rates form an absolute basis for comparison of their economies. However, it should be remembered that a low water rate does not necessarily indicate a low fuel consumption.

Note.—The Foot-pounds Per British Thermal Unit And The British Thermal Units Per Kilowatt-hour Or Per Brake Horsepower-hour are merely different ways of expressing *thermal efficiency* which is discussed in Sec. 245.

242. The Definitions Of The Terms "Total Heat Input" And "Available Heat" should be thoroughly understood

before one attempts to study the significance of the different methods of expressing steam turbine performance values. Consequently, these terms are defined and explained in the following notes:

NOTE.—THE TOTAL HEAT INPUT to the turbine per pound of steam may be defined as the difference between the heat content, $\mathbf{H}_1$, in British thermal units, in 1 lb. of steam at conditions existing at the throttle, and the heat content, $\mathbf{h}_2$, in British thermal units in 1 lb. of water at the temperature of the turbine exhaust. That is,

(44) *Total heat input per lb.* $= (\mathbf{H}_1 - \mathbf{h}_2)$ (B.t.u. per lb.)

Under the steam conditions tabulated in Fig. 245 (150.8 lb. per sq. in., gage, and 100° F. superheat at the throttle, and 92.8° F. at the exhaust), *the total heat input per pound* (Fig. 245) $= \mathbf{H}_1 - \mathbf{h}_2 = 1{,}252 - 61 = 1{,}191$ *B.t.u. per lb.* (The values of $\mathbf{H}_1$ and $\mathbf{h}_2$ are taken from steam tables.) That is, in considering the total heat input per lb., the temperature of the exhaust is taken as the starting or datum point.

NOTE.—THE AVAILABLE HEAT per pound of steam may be defined as the difference between the heat content per pound of the steam under the steam conditions existing at the throttle, $\mathbf{H}_1$, and the heat content per pound of the steam, $\mathbf{H}_2$, after adiabatic expansion to the exhaust pressure. The amount of the "available" heat per pound of steam may be most conveniently obtained by using a steam chart as follows: Find, by the chart (Fig. 15) the heat, $\mathbf{H}_1$, in 1 lb. of steam at the initial conditions. Next, find the heat, $\mathbf{H}_2$, in 1 lb. of steam after adiabatic expansion to the final condition. The difference between these two values is the "available" heat in British thermal units per pound. Expressed as a formula:

(45) *The available heat per lb.* $= \mathbf{H}_1 - \mathbf{H}_2$ (B.t.u. per lb.)

Wherein: $\mathbf{H}_1$ = the heat, in British thermal units, in 1 lb. of steam at the initial steam conditions. $\mathbf{H}_2$ = the heat, in British thermal units, in 1 lb. of steam after it has expanded adiabatically down to the final temperature (at the exhaust). Under the steam conditions outlined in Fig. 245, values being obtained from the steam chart, Fig. 15, *the available heat per lb.* $= \mathbf{H}_1 - \mathbf{H}_2 = 1{,}252 - 894 = 358$ *B.t.u. per lb.*

The reason this "358 B.t.u. per lb." is called the "available" heat for these conditions is because that, with the stated initial and final conditions, it is all of the heat that is available for conversion into mechanical work. It is absolutely all of the heat that could for these conditions be converted into work, even in a theoretically perfect or ideal engine. Why this is true is explained in the author's PRACTICAL HEAT. That is, if a steam engine could be constructed which was an ideal or theoretically perfect engine, it could, under the steam conditions outlined in Fig. 245, convert into work only 358 of the 1,252 B.t.u. per lb. which are supplied to it; the other $(1{,}252 - 358) = 894$ B.t.u. being exhausted.

243. A Rankine-cycle Efficiency value for a certain set of steam conditions indicates the *maximum* percentage of the total heat input (Sec. 242) which a theoretically-perfect ideal vapor engine—steam engine or steam turbine—*could*, when operating between these steam conditions, convert into mechanical work. That is,

(46) $$\textit{Rankine-cycle efficiency} = \frac{\textit{available heat per lb.}}{\textit{total heat input per lb.}} \quad \text{(decimal)}$$

or, using symbols;

(47) $$\textit{Rankine-cycle efficiency} = \frac{H_1 - H_2}{H_1 - h_2} \quad \text{(decimal)}$$

This efficiency is determined solely by the given steam conditions. It constitutes an index of the excellence of the steam conditions. Certain large electric central station companies keep a record of how this efficiency varies from day to day and from month to month for their steam prime movers. Such a record enables the chief engineers to keep check on—and to maintain at maximum effectiveness—the steam conditions under which the prime movers operate. As indicated by For. (46) it is based on the available heat per pound of steam (Sec. 242). Note particularly the example below and the comments which follow it.

Example.—What is the Rankine-cycle efficiency for the steam conditions outlined in Fig. 245? Solution.—By the notes under preceding Sec. 242, *the available heat for the steam conditions of Fig.* 245 *is* 358 *B.t.u. per lb.*, and *the total heat input is* 1,191 *B.t.u. per lb.* Therefore, by (For. 46), the *Rankine-cycle efficiency* = (*available heat*)/(*total heat input*) = 358 ÷ 1,191 = 0.30, or 30 *per cent.* Note that the values used in computing this efficiency are not in any manner dependent upon the operation of the turbine, but only upon the stated initial and final conditions of the steam. Consequently any old kind of a turbine or engine operating under the steam conditions outlined in Fig. 245 would have this same Rankine-cycle efficiency of 30 per cent.

244. A Rankine-cycle Ratio value for a given vapor engine—steam engine or steam turbine—indicates for the given steam conditions, the percentage of the available heat that the given engine converts into mechanical work. It can be determined accurately for a given turbine only by testing the turbine for

work output and observing simultaneously the supply and exhaust steam conditions. Expressed as a formula:

(48) *Rankine-cycle ratio* =

$$\frac{\textit{Work output in B.t.u. per lb.}}{\textit{Available heat per lb.}} \quad \text{(decimal)}$$

or, using symbols;

(49) *Rankine-cycle ratio* =

$$\frac{\textit{Work output in B.t.u. per lb. of steam}}{H_1 - H_2} \quad \text{(decimal)}$$

This efficiency is an index of the excellence of design and mechanical condition of the turbine. Consequently, a comparison of the Rankine-cycle ratios of different vapor engines provides a measure of the excellence of design of the engines for the steam conditions under which each is operating and of its mechanical condition. Thus even though a turbine be excellently designed, if its mechanical condition is permitted to deteriorate—if bearings become scored and blading becomes clogged or broken—its Rankine-cycle ratio will be low. Conversely, a turbine may be well constructed mechanically and be in excellent mechanical condition, but if it is poorly designed its Rankine-cycle ratio will be low.

Explanation.—Consider the turbine, the test results of which are tabulated in Fig. 245. Since from the solution under For. (35), 14.8 *lb. of steam produce* 1 *kw.-hr.*, or 2,655,000 *ft.-lb.*, 1 *lb. of steam produces* (2,655,000 ÷ 14.8) = 179,392 *ft.-lb.* Since there are 778 ft.-lb. in 1 B.t.u., the *number of British thermal units which are, from each pound of steam, converted into work* = 179,392 ÷ 778 = 230 *B.t.u. per lb. of steam.* That is, *the work output is* 230 *B.t.u. per lb. of steam.* This means that in each pound of steam only 230 B.t.u. were actually converted into work; whereas (Sec. 242), there were originally, in each pound, 358 B.t.u. which were *available* for conversion into work. By For. (48), (*the Rankine-cycle ratio*) = (*Work output in B.t.u.*)/(*Available heat*) = 230 ÷ 358 = 0.642 or 64.2 *per cent.* This may be explained as follows: If the turbine had been "perfectly" designed and was in perfect mechanical condition—a theoretically-perfect ideal vapor engine—all of the available 358 B.t.u. per lb. would have been converted into work. But since the turbine only converts 230 B.t.u. per lb. into work, the design and mechanical condition is only 64.2 per cent. "perfect."

245. The Thermal Efficiency expresses the percentage of the *total heat input* of the steam consumed by the turbine which is

converted into work. It is the product of the *Rankine-cycle efficiency* and the *Rankine-cycle ratio.* Thus, it is a sort of an overall efficiency which combines into one value an index of the excellence of the heat conditions (Sec. 243) and of the excellence of the design and mechanical condition (Sec. 244). This combining, into one value, of the expressions for the excellence of heat conditions and of design and mechanical condition may be understood from the following:

(50) *Thermal eff.* = (*Rankine-cycle eff.*) × (*Rankine-cycle ratio*) (decimal)

or, using symbols;

(51) *Thermal eff.* =

$$\frac{H_1 - H_2}{H_1 - h_2} \times \frac{\textit{Work output in B.t.u. per lb. of steam}}{H_1 - H_2} \quad \text{(decimal)}$$

then, simplifying:

(52) *Thermal eff.* =

$$\frac{\textit{Work output in B.t.u. per lb. of steam}}{H_1 - h_2} \quad \text{(decimal)}$$

It is shown in Sec. 243 how the Rankine-cycle efficiency indicates the excellence of heat conditions, and in Sec. 244 how the Rankine-cycle ratio indicates the excellence of design and mechanical condition. Therefore, since the "thermal efficiency formula" (For. 51) contains both of these values, it is evident that the thermal efficiency value must provide an index of the excellence of both heat conditions and design and mechanical condition. Hence the *heat consumption* of turbines of different designs may be intelligently compared on the basis of their thermal efficiencies even when the turbines are operating under different steam conditions. The one which has the highest thermal efficiency will require the least heat for its operation—but the one having the highest thermal efficiency may not be the cheapest to operate because it may cost much more to produce a pound of steam for the steam conditions of the high-thermal-efficiency turbine than it will for the steam conditions of the low-thermal-efficiency turbine; see Div. 14.

Explanation.—Again considering the turbine test results of Fig. 245: From Sec. 243, *the Rankine-cycle efficiency* = 30 *per cent.* From Sec. 244,

the Rankine-cycle ratio = 64.2 *per cent.* By For. (51), *the thermal efficiency* = (*Rankine-cycle efficiency*) × (*Rankine-cycle ratio*) = 0.30 × 0.642 = 0.1926, or 19.3 *per cent.* That is, of the *total heat input per pound of steam* (1,191 B.t.u., Sec. 242), only 30 per cent. (358 B.t.u.) could have, by a theoretically-perfect engine, been converted into work. Furthermore, this particular turbine (Fig. 245) only converted into work 64.2 per cent. of the 30 per cent. which could, under ideal conditions, possibly have been so converted—or, it converted into work only 19.3 per cent., of the total heat input.

246. Graphs Which Show The Total Steam Consumption And The Water Rate Of A Turbine At Various Loads (Fig. 244) are very convenient for comparing (Sec. 249) the operations of two or more turbines; also for comparing test results with the manufacturer's guarantee. Such graphs are obtained as follows: A complete test of the turbine is made at each of the various loads. The total steam consumption and the water rate for each of the several loads are computed as in the preceding example. Then, the total steam consumption, in pounds, and the water rate, in pounds of steam per kilowatt-hour or per brake horsepower-hour, are plotted (Fig. 244) against the load in kilowatts or in brake horsepower. To obtain the data for plotting these curves, tests are usually made at each of the following percentages of full rated load: 50, 75, 100, and sometimes 125 per cent.

247. In Making A Test On A Steam Turbine It is Desirable That Certain Data Be Taken Whereby Any Operating Faults May be Located.—For example, by observing the steam pressure in the various stages (Item 10*c* Sec. 248) information may be obtained as to whether or not the blading is fouled or whether the diaphragm glands are leaking. Also, by comparing the pressure in the exhaust pipe near the turbine with that in the condenser, it will be evident whether or not the pressure drop in the exhaust pipe is excessive. Ordinarily, this pressure drop should not exceed 0.25 to 0.5 lb. per sq. in. Other observations which are not directly essential in determining the performance values (Sec. 240) but which may be used in locating operating faults are tabulated under the following section.

248. A Data Form For A Complete Steam Turbine Test is embodied in the A.S.M.E. Test Code, which is given below:

OBJECT AND PREPARATIONS

Determine the object of the test (Sec. 226), take the dimensions and note the physical conditions not only of the turbine but of the entire plant concerned, examine for leakages, install the testing appliances, etc., as pointed out in the general instructions given in Pars. 1 to 33 (*preceding sections of this division*) and prepare for the test accordingly.

APPARATUS AND INSTRUMENTS

The apparatus and instruments required for a performance test of a steam turbine or turbo-generator, are:

(*a*) Tanks and platform scales for weighing water (or water meters calibrated in place).

(*b*) Graduated scales attached to the water glasses of the boilers.

(*c*) Pressure gages, vacuum gages, and thermometers.

(*d*) Steam calorimeter.

(*e*) Barometer.

(*f*) Tachometer, revolution-counter, or other similar speed-measuring apparatus or equipment.

(*g*) Friction brake or dynamometer.

(*h*) Voltmeters, ammeters, wattmeters, and watt-hour meters for the electrical measurements in the case of a turbo-generator.

Directions regarding the use and calibration of these appliances are given in Pars. 7 to 9, and in Pars. 24 to 33 (A. S. M. E. Test Code, 1915). The determination of the heat and steam consumption of a turbine or turbo-generator should conform to the same methods as those described in the Steam Engine Code, Part V. (*See example under Sec.* 240; also the author's Steam-engine Principles and Practice.) If the steam consumption is determined from the water discharged by the wet vacuum or hot-well pump, correction should be made for water drawn in through the packing glands of the turbine shaft, for condenser leakage, and for any other foreign supply of water.

The rules pertaining to the subjects *Operating Conditions*, *Duration*, *Starting and Stopping*, *Records*, and *Calculation of Results*, are identically the same as those given under the respective headings in the (*A. S. M. E.*) Steam Engine Code, Pars. 71 to 77, with the single exception of the matter relating to indicator diagrams and results computed therefrom; and reference may be made to that code for the directions required in these particulars.

DATA AND RESULTS

The data and results should be reported in accordance with the form (Table 11) given herewith, adding lines for data not provided for, or omitting those not required, as may conform to the object (Sec. 226) in view. If a shorter form of report is desired, the items in fine print designated by letters of the alphabet, may be omitted; or if only the prin-

cipal data and results are desired, the subjoined abbreviated table (Table 12) may be used. Unless otherwise indicated, the items should be the averages of the data.

Table 11. Data And Results Of Steam Turbine Or Turbo-generator Test

Code of 1915

1. Test of ..... turbine located at .....
   To determine .....
   Test conducted by .....
2. Type of turbine (impulse, reaction, or combination) .....
   (*a*) Number of stages .....
   (*b*) Condensing or non-condensing .....
   (*c*) Diameter of rotors .....
   (*d*) Number and type of nozzles .....
   (*e*) Area of nozzles .....
   (*f*) Type of governor .....
3. Class of service (electric, pumping, compressor, etc.) .....
4. Auxiliaries (steam or electric driven) .....
   (*a*) Type and make of condensing equipment .....
   (*b*) Rated capacity of condensing equipment .....
   (*c*) Type of oil pumps (direct or independently driven) .....
   (*d*) Type of exciter (direct or independently driven) .....
   (*e*) Type of ventilating fan, if separately driven .....
5. Rated capacity of turbine .....
   (*a*) Name of builders .....
6. Capacity of generator or other apparatus consuming power of turbine .....

DATE AND DURATION

7. Date .....
8. Duration ..... hr.

AVERAGE PRESSURES AND TEMPERATURES

9. Pressure in steam pipe near throttle by gage ..... lb. per sq. in.
10. Barometric pressure ..... in. of mercury
    (*a*) Pressure at boiler by gage ..... lb. per sq. in.
    (*b*) Pressure in steam chest by gage ..... lb. per sq. in.
    (*c*) Pressure in various stages ..... lb. per sq. in.
11. Pressure in exhaust pipe near turbine, by gage ..... lb. per sq. in.
12. Vacuum in condenser ..... in. of mercury
    (*a*) Corresponding absolute pressure ..... lb. per sq. in.
    (*b*) Absolute pressure in exhaust chamber of turbine ..... lb. per sq. in.
13. Temperature of steam near throttle ..... deg.
    (*a*) Temperature of saturated steam at throttle pressure ..... deg.
    (*b*) Temperature of steam in various stages, if superheated ..... deg.
14. Temperature of steam in exhaust pipe near turbine ..... deg.
    (*a*) Temperature of circulating water entering condenser ..... deg.
    (*b*) Temperature of circulating water leaving condenser ..... deg.
    (*c*) Temperature of air in turbine room ..... deg.

Quality Of Steam

15. Percentage of moisture in steam near throttle, or number of degrees of superheating.....per cent. or deg.
16. Total water fed to boilers.....lb.
17. Total condensate from surface condenser (corrected for condenser leakage and leakage of shaft and pump glands).....lb.
18. Total dry steam consumed (Item 16 or 17 less moisture in steam) .....lb.

Hourly Quantities

19. Total water fed to boilers or drawn from surface condenser per hour.....lb.
20. Total dry steam consumed for all purposes per hour (Item 18 ÷ Item 8).....lb.
21. Steam consumed per hour for all purposes foreign to the turbine (including drips and leakage of plant).....lb.
22. Dry steam consumed by turbine per hour (Item 20 − Item 21).... .....lb.

(*a*) Circulating water supplied to condenser per hour.....lb.

Hourly Heat Data

23. Heat units consumed by turbine per hour [Item 22 × (total heat of steam per pound at pressure of Item 9 less heat in 1 lb. of water at temperature of Item 14)].....B.t.u.

(*a*) Heat converted into work per hour.....B.t.u.
(*b*) Heat rejected to condenser per hour [Item 22*a* × (Item 14*b* − Item 14*a*)] (approximate).....B.t.u
(*c*) Heat rejected in the form of steam withdrawn from the turbine....B.t.u
(*d*) Heat lost by radiation from turbine, and unaccounted for.....B.t.u

Electrical Data

24. Average volts, each phase.....volts
25. Average amperes, each phase.....amperes
26. Average kilowatts, first meter.....kw.
27. Average kilowatts, second meter.....kw.
28. Total kilowatts output.....kw.
29. Power factor.....
30. Kilowatts used for excitation, and for separately driven ventilating fan.....kw.
31. Net kilowatt output.....kw.

Speed

32. Revolutions per minute.....r.p.m.
33. Variation of speed between no load and full load.....r.p.m.
34. Momentary fluctuation of speed on suddenly changing from full load to half-load.....r.p.m.

POWER

35. Brake horsepower, if determined.....b.hp.
36. Electrical horsepower.....e.hp.

ECONOMY RESULTS

37. Dry steam consumed by turbine per b.hp.-hr.....lb.
38. Dry steam consumed per net kw.-hr.....lb.
39. Heat units consumed by turbine per b.hp.-hr. (Item 23 ÷ Item 35) .....B.t.u.
40. Heat units consumed per net kw.-hr.....B.t.u.

EFFICIENCY RESULTS

41. Thermal efficiency of turbine (2,546.5 ÷ Item 39) × 100..... .....per cent.
42. Efficiency of Rankine cycle between temperatures of Items 13 and 14.....per cent.
43. Rankine cycle ratio (Item 41 ÷ Item 42).....

WORK DONE PER HEAT UNIT

44. Net work per B.t.u. consumed by turbine (1,980,000 ÷ Item 39)... ft.-lb.

Table 12. Principal Data And Results Of Turbine Test

1. Dimensions.....
2. Date.....
3. Duration.....hr.
4. Pressure in steam pipe near throttle by gage.....lb. per sq. in.
5. Vacuum in condenser.....in. of mercury
6. Percentage of moisture in steam near throttle or number of degrees of superheating.....per cent. or deg.
7. Net steam consumed per hour.....lb.
8. Revolutions per minute.....r.p.m.
9. Brake horsepower developed.....b.hp.
10. Kw. output.....kw.
11. Steam consumed per b.hp.-hr.....lb.
12. Heat consumed per b.hp.-hr.....B.t.u.
13. Steam consumed per kw.-hr.....lb.
14. Heat consumed per kw.-hr.....B.t.u.

249. A Comparison Of The Performances Of Different Steam Turbines, or of the same turbine at different times, cannot be intelligently made if the computations of the performance values are based on different steam conditions, such as different initial pressures and temperatures, and differ-

ent exhaust pressures and temperatures. Usually, it is impractical to make two tests of the same turbine, or tests of different turbines under the same steam conditions. Consequently, to make a fair comparison between two or more sets of performance values, it is usually necessary to apply certain corrections. Such corrections should be applied which will convert the performance values which are made under one set of steam conditions to those which would obtain under some other set of steam conditions. The amount of the corrections and the method of their application are treated in Div. 13.

QUESTIONS ON DIVISION 12

1. What is the purpose of testing a steam turbine?

2. For what purposes may the *performance values* as computed from the results of a turbine test be used?

3. What should govern the conditions under which a test is made? If the object of the test is to determine how nearly the actual operating performance complies with the guaranteed performance, what are the conditions which should obtain?

4. Name five of the more important data items which should be observed in testing a steam turbine.

5. Over how long a period of time should a turbine test be extended? At what time intervals should the instrument readings be noted and recorded?

6. Why should all instruments used in a turbine test be calibrated both before and after the test?

7. What properties determine the condition of the steam entering the turbine?

8. Explain how the properties which determine the condition of the steam entering the turbine are measured.

9. What property of the steam at the exhaust must be known?

10. Why is it generally desirable to determine the pressure at the exhaust flange of the turbine?

11. What is meant by "referred to a 30-in. barometer?"

12. Name two methods of determining the *power output* of a turbine.

13. What instruments may be used to determine the power output of a turbine electrically? Which instruments are preferable if the load remains constant? If the load fluctuates?

14. In testing a turbo-alternator, why is it desirable that the power factor be unity? What kind of a load will give a unity power factor?

15. How is the *net power output* in watts of a turbo-alternator determined if the exciter is mounted on the generator shaft? If the alternator is excited from a separately driven exciter?

16. Name three methods of determining the quantity of steam consumed by a turbine.

17. Make a sketch of the apparatus required for determining the steam consumption of a turbine which is operated in conjunction with a surface condenser by weighing the condensate.

18. Explain how the *condenser leakage* may be determined. If the condenser leakage is excessive, what should be done before proceeding with the test of the turbine?

19. Make a sketch of the apparatus for determining the *steam consumption* of a turbine by weighing the *boiler feed water*.

20. In determining the steam consumption of a turbine by weighing the boiler feed water, what leakages must be determined? Explain how the amount of this leakage may be measured.

21. Why is it usually undesirable to use a steam flow meter to determine the total steam consumption of a turbine?

22. How may the quantity of the condensate of a jet condenser be measured?

23. How is the speed of rotation of a turbine rotor determined?

24. In what five ways are steam turbine performance values frequently expressed?

25. Explain how each is computed from the test data.

26. Why are the five different methods of expressing a turbine performance included in a test report?

27. Why is the *water rate* used to express turbine performance?

28. For what purpose may the *ft.-lb. per B.t.u.* and the *B.t.u. per kw.-hr.* be used?

29. Define the terms, *total heat input* and *available heat.*

30. What is indicated by a *Rankine cycle efficiency* value?

31. What is indicated by a *Rankine cycle ratio* value? To what is it an index?

32. What is indicated by the *thermal efficiency?* To what is it an index?

33. For what purposes may graphs, which show the *total steam consumption* and the *water rate* of a turbine at various loads, be conveniently used? Explain how such graphs may be obtained.

34. If two or more turbines have been tested under different steam conditions, what must be done before their performance values can be intelligently compared?

35. Make a sketch showing location of all instruments used in testing a turbo-alternator which is operated in conjunction with a surface condenser.

DIVISION 13

EFFECT OF STEAM PRESSURE, SUPERHEAT, AND VACUUM ON STEAM-TURBINE ECONOMY

250. The Water Rate And Thermal Efficiency Of A Turbine Are Dependent On The Conditions Of The Supply And Exhaust Steam.—In general, it may be said that the greater is the heat content of the supplied steam and the smaller is the heat content of the exhaust steam, the higher will be the thermal efficiency of the turbine and the lower will be its water rate. Hence, those factors which produce great heat content in steam—high pressure, high quality, and high superheat—are to be desired as properties of the supply steam. Also, those factors which produce small heat content in the exhaust steam—low exhaust pressure (high vacuum) and little steam friction and leakage within the turbine—are very desirable. Unfortunately, however, it always costs more to produce supply steam of great heat content—high pressure and superheat—than it does to produce supply steam of small heat content. Likewise, the condensers, cooling water, and auxiliary power for high-vacuum service cost more than for low-vacuum service. Hence, it is the object of this division to study the several effects of the above steam conditions on the efficiency of turbines and on their cost of operation so that the most economical conditions for any given turbine may be determined. Figs. 245*A* and 245*B* illustrate the steam conditions in a large turbine.

NOTE.—THE EFFECTS OF PRESSURE, SUPERHEAT, AND VACUUM ON THE WATER RATE AND THERMAL EFFICIENCY OF A THEORETICALLY PERFECT TURBINE will first be discussed because the effects in a theoretically perfect turbine are explanatory of the effects in an actual or commercial turbine. Wherever the effects in an actual turbine are different from those in the theoretical, these differences will be explained at a later point in this text.

EXPLANATION.—The water rate of a theoretically perfect turbine is given in For. (19) which is restated below as For. (53). The thermal efficiency of a theoretically perfect turbine, which is the same as its

Rankinecycle efficiency,. is given by For. (54) which is derived in the author's PRACTICAL HEAT.

$$W_H = \frac{2{,}545}{H_1 - H_2} \quad \text{(lb. per hp.-hr.)} \tag{53}$$

$$E_T = \frac{H_1 - H_2}{H_1 - h_2} \quad \text{(decimal)} \tag{54}$$

Wherein: W_H = the turbine water rate, in pounds per horsepower-hour. E_T = the turbine's thermal efficiency, expressed decimally. H_1 = the total heat of 1 lb. of supply steam, in British thermal units. H_2 = the total of 1 lb. of steam after adiabatic expansion to the exhaust pressure, in British thermal units. h_2 = the heat of 1 lb. of water at the temperature which is the boiling point at the exhaust pressure, in British thermal units. H_1 and H_2 may, as explained in Div. 1, be found from a total-heat-entropy chart (Fig. 15); h_2 is found from the steam tables.

Inspection of For. (53) shows that the greater is the difference between H_1 and H_2, for a given turbine, the smaller will be the water rate of the turbine. Hence, changes in the steam conditions which increase H_1 or which decrease H_2, will enable the turbine to operate with a lower water rate—and *vice-versa*. With regard to For. (54), however, it is not evident by inspection just what effects on the thermal efficiency will be produced by changes in the steam conditions. To illustrate the effects of changing the quality, pressure, and superheat of the supply steam and of changing the exhaust pressure (vacuum), the specific examples following Table 251 are here given.

251. Table Showing The Effect Of Different Steam Conditions On The Water Rate And Thermal Efficiency Of A Theoretically Perfect Steam Turbine.—The method of computing these values is shown in the following examples.

Condition number	Supplied steam				Exhaust steam			Water rate, lb. per hp.-hr.	Thermal or Rankine-cycle efficiency, per cent.	Decrease in water rate, per cent.	Increase in efficiency, per cent.
	Pressure, lb. per sq. -in. gage	Quality, per cent.	Superheat, degrees Fahr.	H_1	Vacuum, inches mercury gage	H_2	h_2				
1	150	90	...	1,110	28	813	94	8.58	29.2		
2	150	100	0	1,195	28	870	94	7.84	29.5	8.6	1.0
3	175	100	0	1,198	28	863	94	7.60	30.3	3.0	2.7
4	175	...	150	1,282	28	917	94	6.98	30.7	8.1	1.3
5	175	...	150	1,282	29	883	70	6.38	32.9	8.6	7.1

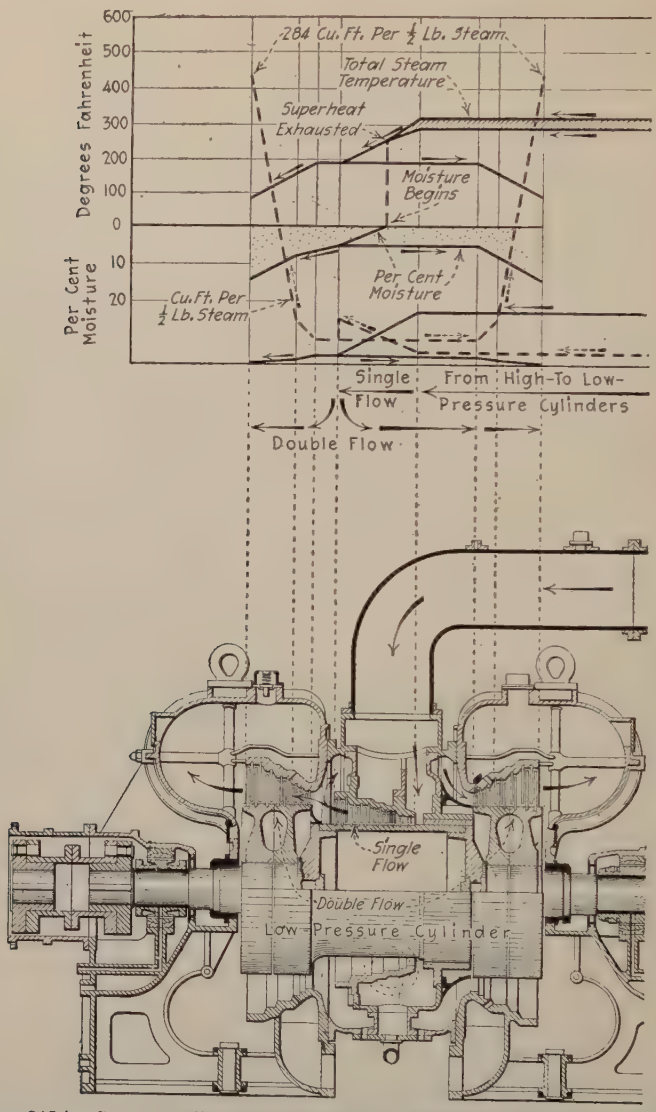


Fig. 245A.—Steam conditions in a 40,000 kw. Westinghouse turbine when it is delivering 28,000 kw. The primary valve is admitting steam at 250 lb. per sq. in. The secondary valve is just beginning to open, The tertiary valve is closed. (*Power*, August 8, 1922.)

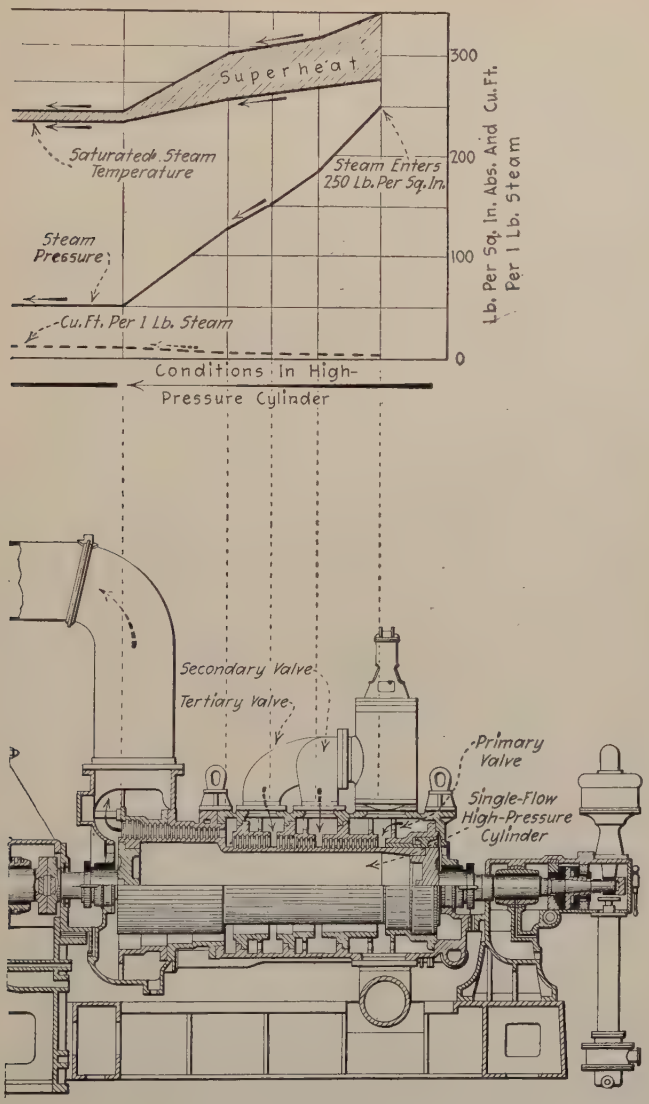


Fig. 245*B*.—Continuation of Fig. 245*A*.

Example.—First Condition.—*Supplied steam pressure*, 150 *lb. per sq. in. gage; quality*, 90 *per cent.; vacuum*, 28-*in. mercury column.* What are the water rate and the efficiency of this perfect turbine? Solution.—From the total-heat-entropy chart of Fig. 15, the value of $\mathbf{H}_1$ is found at the intersection of the 90 per cent. quality line and the 150-lb. pressure line to be 1,100 *B.t.u.* Following (on Fig. 15) vertically downward to the 28-in. vacuum line, $\mathbf{H}_2$ is found to be 813 *B.t.u.* From a steam table, $\mathbf{h}_2$ is found to be 94 *B.t.u.* Hence, by For. (53): *the water rate* $= \mathbf{W}_H = 2{,}545/(\mathbf{H}_1 - \mathbf{H}_2) = 2{,}545 \div (1{,}100 - 813) = 8.58$ *lb. per hp.-hr.* This result could also have been read from the scale A at the top of Fig. 15. Also, by For. (54): *The thermal efficiency* $= \mathbf{E}_T = (\mathbf{H}_1 - \mathbf{H}_2)/(\mathbf{H}_1 - \mathbf{h}_2) = (1{,}100 - 813) \div (1{,}100 - 94) = 0.292$ or 29.2 *per cent.*

Example.—Second Condition.—*Supplied steam pressure*, 150 *lb. per sq. in. gage; quality, dry saturated; vacuum*, 28-*in. mercury column.* What are the water rate and thermal efficiency of the turbine under these conditions and how much have they been improved? Solution.—In the same manner as in the first condition, *the water rate* is found to be 7.84 *lb. per hp.-hr.*, and the *thermal efficiency* to be 29.5 *per cent.* Hence, the *decrease in water rate* $= 8.58 - 7.84 = 0.74$ *lb.* or $(0.74 \div 8.58) = 0.086$ or 8.6 *per cent.* Also, the *increase in efficiency* $= 29.5 - 29.2 = 0.3$ *per cent.*, or an *improvement* of $(0.3 \div 29.2) = 0.01 = 1$ *per cent.*

Example.—Third Condition.—*Supplied steam pressure*, 175 *lb. per sq. in. gage; quality, dry saturated; vacuum*, 28-*in. mercury column.* What are the water rate and thermal efficiency of the turbine under these conditions and how much have they been improved? Solution.—In the same manner as for the first condition, *the water rate* is found to be 7.60 *lb. per hp.-hr.*, and *the thermal efficiency* to be 30.3 *per cent.* Hence, *the decrease in water* $= 7.84 - 7.60 = 0.24$ *lb.* or $(0.24 \div 7.84) = 0.03$ or 3 *per cent.* Also, *the increase in efficiency* $= 30.3 - 29.5 = 0.8$ *per cent.*, or an *improvement* of $(0.8 \div 29.5) = 0.027$ or 2.7 *per cent.*

Example.—Fourth Condition.—*Supplied steam pressure*, 175 *lb. per sq. in. gage; superheat*, 150° *F.; vacuum*, 28-*in. mercury column.* What are the water rate and thermal efficiency of the turbine under these conditions and how much have they been improved? Solution.—In the same manner as for the first condition, *the water rate* is found to be 6.98 *lb. per hp.-hr.*, and *the thermal efficiency* to be 30.7 *per cent.* Hence, *the decrease in water rate* $= 7.60 - 6.98 = 0.62$ *lb.* or $(0.62 \div 7.60) = 0.081$ or 8.1 *per cent.* Also, *the increase in efficiency* $= 30.7 - 30.3 = 0.4$ *per cent.*, or an *improvement* of $(0.4 \div 30.3) = 0.013$ or 1.3 *per cent.*

Example.—Fifth Condition.—*Supplied steam pressure*, 175 *lb. per sq. in. gage; superheat*, 150 °*F.; vacuum*, 29-*in. mercury column.* What are the water rate and thermal efficiency of the turbine under these conditions and how much have they been improved? Solution.—In the same manner as for the first conditions, *the water rate* is found to be 6.38 *lb. per hp.-hr.*, and *the thermal efficiency* to be 32.9 *per cent.* Hence, *the decrease in water rate* $= 6.98 - 6.38 = 0.60$ *lb.* or $(0.60 \div 6.98) = 0.086$ or 8.6 *per cent.* Also, *the increase in efficiency* $= 32.9 - 30.7 = 2.2$ *per cent.*, or an *improvement* of $(2.2 \div 30.7) = 0.071$ or 7.1 per cent.

252. Theoretically, The Water Rate And Thermal Efficiency Of A Turbine Depend Only On The State Of The Supply Steam And On The Exhaust Pressure Or Vacuum.—How the initial steam pressure and quality or superheat and the vacuum affect the water rate and efficiency is shown by the preceding typical examples. It is to be noted from the examples and from Table 251 that the increase in efficiency with changed conditions is not of the same magnitude as is the decrease in water rate. These examples show clearly that the water rate alone should not be taken as a measure of a turbine's thermal performance—as a measure of the fuel that must be consumed to insure its operation.

253. Actually, The Water Rate And Thermal Efficiency Depend Also On The Amount Of The Losses Within The Turbine.—As stated in Sec. 15, losses occur within a turbine casing due to several causes, the principal ones being steam friction, steam leakage, eddy currents, radiation, and the velocity of the exhaust steam. All of these losses except that due to radiation tend to increase the value of H_2 in Fors. (53) and (54); hence, they tend to increase the water rate and decrease the efficiency. Furthermore, all of these losses are dependent on the quality of the steam in the various passages of the turbine (as explained further hereinafter)—the drier the steam, the less are the losses. Now, in any turbine, the quality of the steam decreases rapidly as the steam flows through its passages. Hence, any change, so made in the condition of the supply steam as to increase the quality of the steam in the turbine passages, is certain to reduce the losses within the turbine and to thereby decrease its water rate and increase its thermal efficiency.

Note.—The Percentage Losses Are Greater In Turbines Of Small Than In Those Of Large Capacity.—That this is true is shown by the variation of the efficiency ratio, E_r, in Fig. 20. The explanation of the variation in losses lies in the fact that the interior-surface areas of a turbine and the places of possible leakage are greater (in proportion to the amount of steam used by the turbine) in small-capacity turbines than in large-capacity turbines.

254. Every Turbine Is Designed For Specific Steam Conditions and will perform most efficiently when operated

under those conditions. The actual conditions under which a turbine will operate most efficiently may or may not be the same as those conditions for which the turbine was furnished by its manufacturer and which are stamped on its name plate; this is because some shop standardization is necessary in turbine building and each turbine cannot be specially designed for the purchaser. It follows that, in general, a turbine should always be operated under the steam conditions for which it was designed. Hence, the efficiency of turbines will not always be increased by increasing the supplied steam pressure, superheat, or the vacuum. In fact, if too great a departure is made from the conditions for which the turbine is designed, the efficiency may be decreased instead of increased, as explained below. Hence, the manufacturer of a turbine should always be consulted as to the effects of condition changes before any material changes are made. The manufacturer can advise definitely as to whether or not your contemplated change is feasible and also as to the economies which will thereby be effected.

Explanation.—Why A Turbine Should Be Operated Only Under The Steam Conditions For Which It Was Designed may be explained thus: Any change in the steam conditions will, as shown below, increase the losses in the turbine. If the steam pressure at the throttle is increased and the amount of the superheat and vacuum are unchanged, or if the vacuum is increased and the amount of superheat and pressure are unchanged, the pressure range of the turbine, or the pressure drop through it, is increased. Consequently the pressure drop in each stage is increased causing the steam to strike the blades with a greater velocity than that for which they were designed. Any change in the value of this velocity causes the steam to hit the blades at an angle instead of tangentially thereby increasing the loss due to impact. There is also a loss due to the increase in the amount of moisture in the steam near the exhaust but this loss also occurs in a turbine designed for the improved conditions. Increasing the vacuum has the further disadvantage of increasing the volume of the exhaust steam. This means that the velocity of the steam in the passages near the exhaust end of the turbine must be increased, and produces a loss due to exit velocity and to the increased friction.

An increase in the amount of superheat, with the amount of the vacuum and pressure unchanged, increases the volume of the steam that must pass through the turbine per unit of time. The only manner in which this greater volume of steam can be forced through the passages is by increased velocity. A greater velocity means larger friction and

impact losses. The capacity of the turbine may even be reduced if the amount of superheat is increased too much. The resulting losses may then offset, to a greater or less degree, the increase in efficiency which the improved steam conditions should theoretically provide.

255. The Capacity Of Any Existing Turbine May Be Increased By Increasing The Supply Pressure, Superheat, And Vacuum—any one, two or all three. But while the capacity of the machine will be thereby increased, it will usually be at the expense of efficiency. Just what will be the effect on economy of such an increase in capacity is determined by the design and construction of the turbine. The manufacturer can furnish exact information.

256. Table Showing Factors For Computing The Approximate Change In The Water Rate Of A Turbine With Changed Steam-supply Pressure, Superheat, And Vacuum.—The application of these factors is explained and illustrated in following

Change in steam condition		Change in water rate
SUPPLY-STEAM PRESSURE. (Increasing the supply-steam pressure **decreases** the water rate and vice versa.)		Turbines up to 1,000 kw.—1.5 per cent. for each 10 lb. per sq. in. change in pressure
		Turbines over 1,000 kw.—1.0 per cent. for each 10 lb. per sq. in. change in pressure.
SUPPLY-STEAM SUPERHEAT. (Increasing the superheat **decreases** the water rate and vice versa.)		Up to 100° F. superheat—1.0 per cent. for each 10° F. of change in superheat.
		100° to 150° F. superheat—0.8 per cent. for each 10° F. of change in superheat.
		150° F. to 250° F. superheat—0.6 per cent. for each 10° F. of change in superheat.
EXHAUST PRESSURE	Back pressure. (Increasing back pressure **increases** the water rate and vice versa.)	Up to 15 lb. per sq. in. gage—2 to 3.5 per cent. for each pound of back-pressure increase (see Fig. 251).
	Vacuum. (Increasing the vacuum **decreases** the water rate and vice versa.)	Between 25 and 27 in.— 5 per cent. per inch of vacuum.
		Between 27 and 28 in.—6 per cent. per inch of vacuum.
		Between 28 and 29 in.—10 per cent. per inch of vacuum.

sections. The preceding factors are approximately correct for condition changes within reasonable limits, whether or not the turbine is changed to suit the new conditions; see note below.

Note.—Exact Values Indicating The Effects Of Changing Steam Conditions cannot be given because the exact values depend on the design and construction of the turbine under consideration and upon the steam conditions—pressure, superheat, and vacuum—prior to changing the steam conditions. For any given turbine, exact factors, in the form of graphs similar to Figs. 252, 253, and 254, may be obtained from the manufacturer.

257. Turbines Are More Efficient When, Other Things Being Equal, They Are Supplied With Steam At High Pressure.—As suggested by Fig. 246 and also by Fig. 15, the higher the pressure of steam, the more heat per pound there is in it. That is, the higher the pressure, the greater will be the value of H_1 in Fors. (53) and (54). The greater the value of H_1—other things being equal—the smaller will be the water rate and (generally) the greater will be the thermal efficiency. This is brought out by Table 251 and by the examples which follow it. But if the turbine is to efficiently use the high-pressure steam, it must have been designed (Sec. 254) for that pressure.

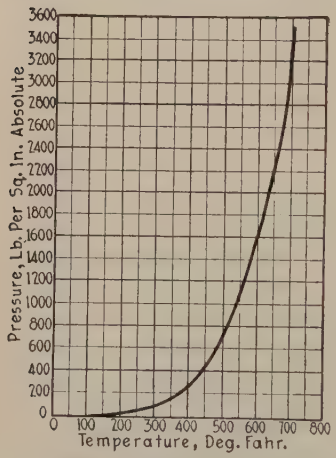


Fig. 246.—Graph showing the variation of the temperature of saturated steam with the steam pressure.

258. The Effect Of Increasing The Supply-steam Pressure Of An Existing Turbine is generally to increase, to some extent, the efficiency of the turbine. But, the turbine may require new nozzles for the higher pressure and, if the turbine is already operating on steam at a pressure near that for which it is designed, or if the turbine is operated most of the time at fractional loads, the efficiency may be increased but slightly

or it even may be decreased by increasing the supply pressure. Furthermore, steam at higher pressures costs more to produce than does steam at lower pressures—the boiler losses are greater, and more expensive boilers must be used. Hence, to determine whether a change to a considerably higher steam-pressure is advisable, it is best to consult the turbine builder's engineering department.

NOTE.—THE STEAM PRESSURES WHICH ARE ADVISABLE FOR TURBINE OPERATION are as follows: For small turbines, say up to 200-kw. capacity, about 150 to 175 lb. per sq. in. gage. For medium-capacity turbines, say 200 to 5,000 kw., about 200 to 250 lb. per sq. in. gage. For large capacity turbines, as in the large central stations the tendency is continually toward higher pressures—some now use pressures as high as 350 lb. per sq. in. gage. It is doubtful whether pressures higher than 400 lb. will be used, however, because of the high cost and maintenance expense of boilers for these high pressures and because the thermal gains from further pressure increase are very small. Note from Fig. 246 that the steam temperature—which determines, somewhat, the value of $\mathbf{H}_1$ in For. (53)—increases very slowly with the pressure for pressures exceeding 400 lb. per sq. in.

259. To Compute The Effect On A Turbine's Water Rate Of Changing Its Supply Pressure, the factors given in Table 256 may be used whenever manufacturers' correction curves (Sec. 268) are not obtainable. The factors in Table 256 are to be used only for computing the effect of changes which do not exceed 10 to 15 per cent. of the rated steam pressure.

EXAMPLE.—The rated steam-supply pressure for a 2,000-kw. turbo-generator is 175 lb. per sq. in. gage (the superheat and vacuum may, within reason, be any whatsoever). The water rate of the machine is 17 lb. per kw.-hr. What water rate may be expected if the steam pressure is raised from 175 to 200 lb. per sq. in. gage? SOLUTION.—The increase in pressure is: 200–175 = 25 lb. per sq. in. Now, $25 \div 10 = 2.5$. Since, from Table 256, a 1 per cent. decrease in water rate may be expected for each 10 lb. per sq. in. pressure increase, the decrease in this case will be 2.5 per cent. Hence, the *water-rate decrease* $= 0.025 \times 17 = 0.43$ *lb. per kw.-hr.* Therefore, at 200 lb, per sq. in. pressure, the *water rate* $= 17 - 0.43 = 16.57$ *lb. per kw.-hr.*

260. Turbines Are More Efficient When, Other Things Being Equal, They Are Supplied With Highly Superheated Steam.—For a given pressure, the value of $\mathbf{H}_1$, Fors. (53) and

(54), increases with the superheat. Hence the water rate decreases with the superheat and (usually) the thermal efficiency increases. In using high superheat, however, one must be careful that the superheat is not so high that it causes the exhaust steam to be superheated—this would result in a loss. In general, it may be said that high pressures with moderate superheat are more economical than moderate pressures with high superheats.

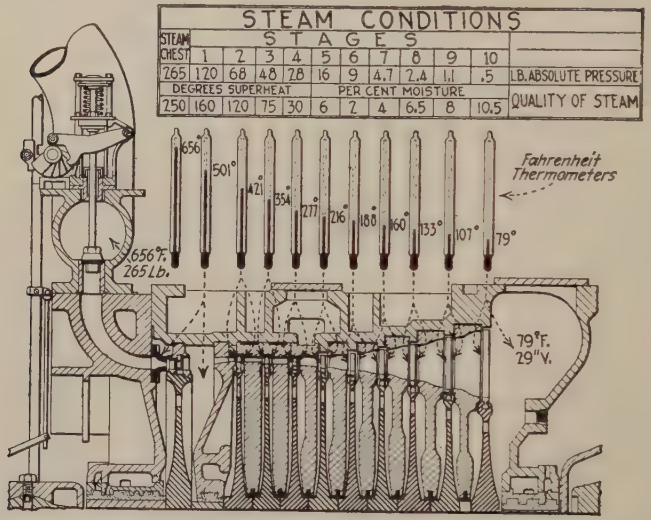


FIG. 246*A*.—Showing the condition of the steam in each stage of a ten-stage turbine. (*General Electric Review*, March, 1918.)

Steam turbines are, inherently, exceedingly well adapted to the economic use of superheated steam.

NOTE.—THE REASON WHY SUPERHEATING ITS SUPPLY STEAM IMPROVES THE ECONOMY OF A TURBINE is that the superheating reduces the water-vapor friction in the turbine: As steam expands in passing through successive stages in a turbine and gives up heat which makes the rotor turn, the quality of the steam tends to become reduced (Fig. 246*A*). That is, the steam tends to condense and produce water vapor—minute drops of water in suspension. The friction of the turbine's rotating disks in such a dense wet vapor is considerably higher than in dry steam. Similarly the friction of "wet" steam passing through the nozzles and buckets is greater than that of dry steam. This friction represents wasted energy. The higher the *quality* of the steam the less the friction. The more the supply steam is superheated the further it will travel

through (the more heat it can give up in) a given turbine without condensation—without its becoming saturated. Hence, even a little superheating, of the supply steam for a turbine is very valuable. Also, superheating has the added advantage of minimizing blade and nozzle erosion in a turbine.

261. The Effect Of Increasing The Supply-steam Superheat Of An Existing Turbine is generally to increase, to some extent, the efficiency of the turbine. The principal effect of increasing the superheat is to decrease the amount of moisture (water) in the steam in the several passages of the turbine (see preceding note); hence, superheating decreases the amount of the losses within the turbine. The principal objection to the use of highly

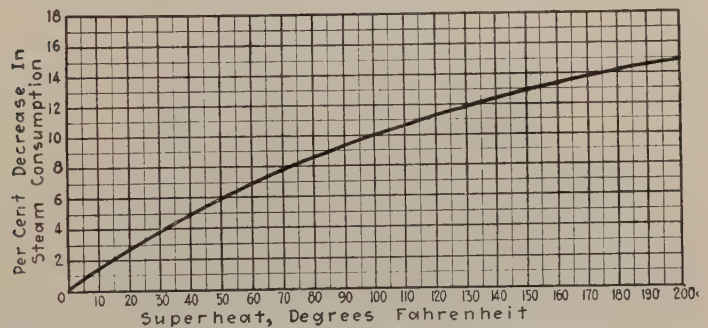


Fig. 247.—Graph showing the effect of superheat on steam consumption of non-condensing steam turbines. (*Sturtevant Company.*)

superheated steam is that, especially in some types of turbines (those which have many rows of blades), the high-pressure end of the turbine becomes heated to such a high temperature that the casing is severely strained. Because turbines have no rubbing surfaces which are exposed to the high-pressure steam (as have steam engines), there are no lubrication difficulties occasioned by the use of superheated steam in turbines. In any case, however, the cost of superheating the steam (see Fig. 248, which is explained hereinafter) must be balanced against the gain in efficiency which is produced. The net economic value of superheating is thus determined.

Note.—The Superheats Which Are Most Advisable For Turbine Operation may roughly be taken as two-thirds of the steam-supply pressure in pounds per square inch gage. Thus, about 125° to 150° F. of

superheat is advisable for medium-sized plants whereas superheats of about 200° F. are used in large central stations. Furthermore, non-condensing turbines generally require more superheat and are benefited more thereby than condensing turbines. However, unusual local conditions such as very-high or very-low fuel cost may render the above values inapplicable. Each case should be considered individually on its merits. The effect of superheat on a non-condensing turbine is shown in Fig. 247.

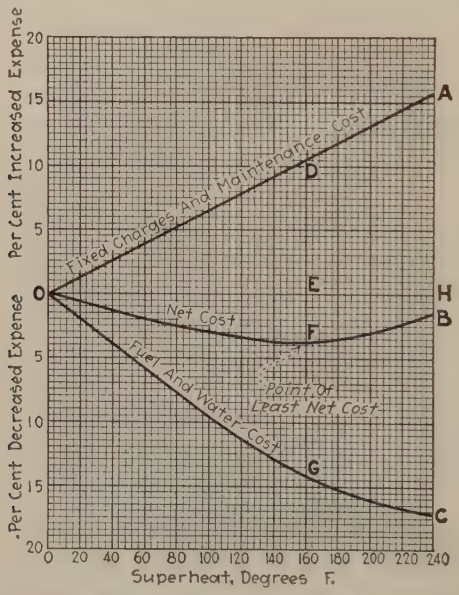


Fig. 248.—Showing typical relation of power-production cost to superheat. This graph is plotted for certain conditions (175 lb. per sq. in. pressure and 210° F. feed water in a certain plant) but the general principle which it illustrates is characteristic. For these conditions, the greatest decrease in net cost at *F*, due to superheating occurs with a superheat of 160° F. The net decrease in cost, *EF* = (*Decrease in fuel and water cost*, *EG*) − (*The increase fixed charge and maintenance cost*, *ED*). That is, to determine the locations of the points along in *OB*, for each different superheat, the corresponding vertical distance between *OC* and *OH* is laid off vertically downward, that is subtracted, from *OA*.

262. The Actual Net Fuel Saving Due To Superheating A Turbine's Supply Steam is usually about 2 to 5 per cent. per 100° F. of superheat increase, the superheating to be within practical limits. Excessive superheating is not economical (Fig. 248) because the increased cost of the fuel required and the additional expense of equipment for producing and transmitting the superheated steam, more than offsets the decreased

fuel consumption due to its use. Advisable superheats are given in the preceding note.

Example.—In the plant and for the conditions for which Fig. 248 was plotted, the most economical superheat (at *F*) is 160° F. With this superheat the net cost of power production is 4 per cent. less than if no superheat were employed. With less superheat than 160° F., as shown by *FO*, or with more superheat than 160° F., as shown by *FB*, the net cost of power is greater.

263. To Compute The Effect On A Turbine's Water Rate Of Changing The Superheat, the factors given in Table 256 may be used whenever manufacturer's correction curves (Sec. 268) are not obtainable. The method of computing the effects of superheat changes is illustrated by the following example.

Example.—A certain turbine (the supply-steam pressure and the vacuum may be any within reason) shows a water rate at full load of 14 lb. per hp.-hr. when supplied with steam of 50° F. superheat. What would be its water rate if the superheat were raised to 150° F.? Solution.—By Table 256 each 10° F. of superheat increase between 0° F. and 100° F. decreases the water rate by 1 per cent., and each 10° F. of superheat increase between 100° F. and 150° F. decreases the water rate by 0.8 per cent. Hence, for this turbine, the *percentage decrease in water rate* = $[(100 - 50) \times 1] \div 10 + [(150 - 100) \times 0.8] \div 10 = 5 + 4 = 9$ *per cent.* Hence, the *pounds decrease in water rate* $= 0.09 \times 14 = 1.26$ *lb. per hp.-hr.* Therefore, the *water rate with* 150° F. *superheat* $= 14 - 1.26 = 12.74$ *lb. per hp.-hr.*

264. High Vacuum Is The Most Essential Requirement For Economical Steam-turbine Operation; see Table 251 and the examples which follow it. Maintaining a high vacuum provides the most effective method of insuring good economy of condensing turbines. Condensing turbines are, in general, more economical—often much more; see Div. 14—than are condensing reciprocating engines, principally because the turbine is inherently better adapted to the use of high vacuums; see below. As a general rule, it pays to keep the vacuum in a turbine's exhaust pipe at as high a value as the plant conditions and water supply will permit. However, it may not always pay to circulate all the water which the condenser pumps can

handle—above a certain vacuum the cost of pumping additional water may be greater than the fuel saving due thereto. This is particularly true when the turbine is operating under partial load, or in winter when the circulating water is very cold. Here again, each turbine is deserving of a separate economic study to determine the most economical vacuum in different seasons.

Note.—Turbines Can Effectively Utilize Higher Vacuums Than Engines For Two Reasons: (1) *Turbine parts are always subjected to steam at the same pressure;* the low temperatures of the exhaust pressure cannot reach back into the hotter parts of the machine whereas in steam engines the cylinders are exposed alternately to wide differences of temperature—this causes cylinder condensation. (2) *The steam expansion is not limited in the turbine* whereas, in the engine, expansion is limited by

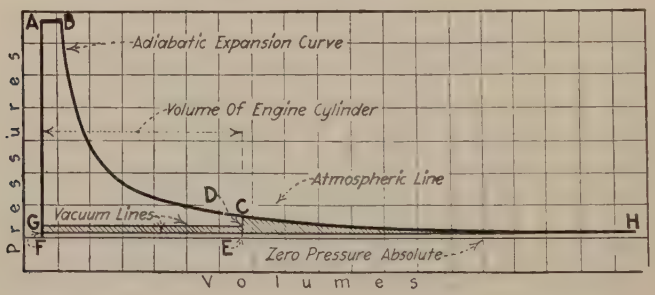


Fig. 249.—Pressure-volume curves for steam engines and turbines.

the cylinder volume. (If, with a steam engine, an effort is made to provide very-great cylinder volume, the low-pressure cylinder will then become excessively large. The friction and other losses which the very large cylinder would introduce, much more than offset the economies which would theoretically result from the increased cylinder volume.) Take, as an example, a condensing steam engine which has an ideal diagram as shown in Fig. 249 at *ABCDG*. Since the expansion is limited by the cylinder volume, a higher vacuum will increase the diagram area only by the strip *FGDE*. Since, within a turbine, expansion can be carried down to the lowest condenser pressures, turbines can utilize the large triangular expansion area, *CHE*, in addition to all that the engine gains. See the author's Steam Power Plant Auxiliaries And Accessories for further information.

265. The Usual Vacuums Carried In Steam-turbine Practice Are as follows: (1) *Where circulating water is not*

very plentiful or where it must be pumped great distances: 27 to 28 in. (2) *Where circulating water is plentiful and always in large-capacity stations:* 28 to 29 in. The smaller values are, in each case, the vacuums carried in the summer months; the lower values are those which are carried in the winter months. Although the values given above are quite commonly observed, the most economical vacuum should be determined for every plant before adopting a standard. This is done by a comparison of operating costs with different vacuums. Higher average vacuums, and consequently more economical operation,

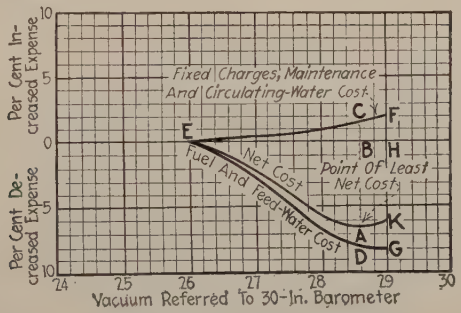


FIG. 250.—Showing typical relations of power-production cost to vacuum. This graph is plotted for specific conditions but the general principle which it illustrates is characteristic. For these conditions, the greatest decrease in net cost, at *A*, occurs with a vacuum of 28.6 in. mercury column. *The net decrease in cost, BA, = (Decrease in fuel and feed-water cost, BD) − (Increase in fixed charges, maintenance, and circulating-water cost, BC).* That is, to determine the locations of points along *EK* for each different vacuum, the corresponding vertical distance between *EF* and *EH* is subtracted from the vertical distance between *EG* and *EH*.

are always possible in the northern than in the central and southern states. This is because of the lower temperatures of the cooling water in the northern states.

EXAMPLE.—In the plant and for the conditions for which Fig. 250 was plotted, the most economical vacuum (at *A*) is 28.6 in. of mercury. With this vacuum the net cost of power production is 6.6 per cent. less than if only a 26-in. vacuum were carried. With more or less vacuum than 28.6 in., the net power cost would be greater, as shown by *AK* and *EA*.

266. To Compute The Effect On A Turbine's Water Rate Of Changing The Vacuum, the factors given in Table 256 may be used whenever manufacturer's correction curves (Sec. 268)

are not available. The values given in Table 256 are applicable only up to the vacuum at which the turbine was designed to be most efficient—generally 28.5 to 29 in. The method of applying these factors is illustrated in the following example.

Example.—A certain turbine, when operating under a 27-in. vacuum, has a water rate of 12 lb. per kw.-hr. (The supply-steam pressure and the superheat—if any—may be any reasonable values.) What water rate may be expected from this turbine when operating under a 28.5-in. vacuum? Solution.—By Table 256 the water rate will be decreased 6 per cent. by raising the vacuum to 28 in., and will be further decreased $0.5 \times 10 = 5$ per cent. by raising the vacuum from 28 to 28.5 in. Hence, the *per cent. decrease* $= 6 + 5 = 11$ *per cent.* Therefore, the *water rate decrease* $= 0.11 \times 12 = 1.31$ *lb.*, or, with a 28.5-in. vacuum, the *actual water rate* $= 12 - 1.31 = 10.69$ *lb. per kw.-hr.*

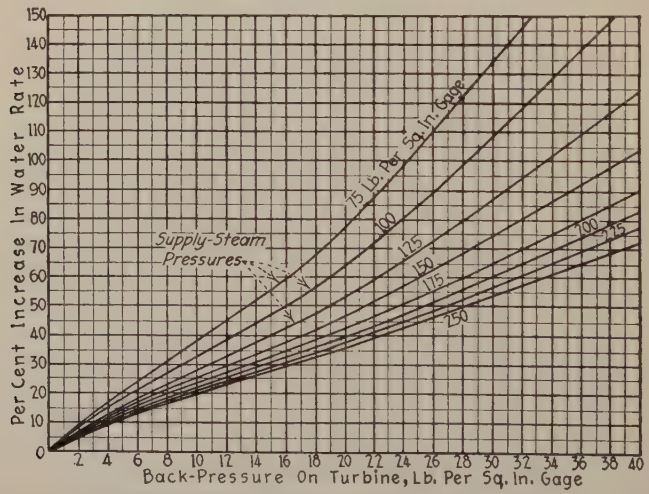


Fig. 251.—Graphs showing effects of increasing the back pressure on the water rates of non-condensing turbines. (*B. F. Sturtevant Co.*)

267. Increasing The Back Pressure On A Non-condensing Turbine Increases The Water Rate And Decreases The Thermal Efficiency (Fig. 251).—Since the back pressure on a non-condensing turbine corresponds exactly to the vacuum on a condensing turbine, all of the previous discussion concerning vacuums applies, to a greater or less degree, to back-pressures—the chief difference being in the magnitude of the effect of a given pressure change in the two cases. The graphs

of Fig. 251 illustrate the effects on the water rates of increasing the back-pressure from atmospheric to different values and shows how these effects vary with different initial (supply) steam pressures.

Note.—To Compute The Effect On A Non-condensing Turbine's Water Rate Of Changing The Back Pressure, the factors given in Table 256 or the graphs of Fig. 251 (which are more accurate) may be used. The method of using these graphs is illustrated in the following example.

Example.—A non-condensing turbine which is operating with a supply pressure of 150 lb. per sq. in. gage (any reasonable superheat or no superheat) and a back pressure of 10 lb. per sq. in. gage, shows a water rate, by test, of 44.8 lb. per hp.-hr. What water rate might the turbine be expected to have if the back pressure were raised to 25 lb. per sq. in. gage? Solution.—From Fig. 251, the water rate with 10-lb. back pressure and 150 lb. per sq. in. supply pressure, is 25.5 per cent. higher than it would be with atmospheric exhaust. Hence, with atmospheric exhaust, *the water rate* $= 44.8 \div (1.00 + 0.255) = 35.7$ *lb. per hp.-hr.* Also, from Fig. 251, the water rate with 25-lb. back pressure is 60 per cent. higher than with atmospheric exhaust. Hence, with 25-lb. back pressure, *the water rate* $= 35.7 + (0.60 \times 35.7) = 35.7 + 21.4 = 57.1$ *lb. per hp.-hr.*

268. Manufacturers Sometimes Supply Performance "Correction Graphs" With Turbines (Figs. 252, 253 and 254). The purpose of such graphs is to provide the purchaser with more accurate means, than the factors of Table 256, for computing the probable effects on the turbine's water rate of changing the supply pressure, superheat, and vacuum. A very important application of such curves is for making "corrections" to the results of an acceptance test (Sec. 226) in which the exact steam conditions of the manufacturer's guarantee did not prevail. The use of performance correction graphs for verifying guarantees is explained in following Sec. 269.

269. The Water Rates At The Steam Conditions Of An Acceptance Test May Be Corrected To The Water Rates Which Would Have Obtained If The Acceptance Test Had Been Made Under The Steam Conditions Of The Guarantee by the following formulas:

$$(55) \qquad C = \left(1 - \frac{W_G}{W_P}\right) + \left(1 - \frac{W_G}{W_S}\right) + \left(1 - \frac{W_G}{W_V}\right) \text{ (decimal)}$$

$$(56) \qquad W_C = W_T - CW_T \text{ (lb. per kw.-hr.)}$$

Wherein: C = the net correction factor, expressed as a decimal. $\mathbf{W}_G$ = the full-load water rate of the turbine at the steam conditions specified in the guarantee. $\mathbf{W}_P$ = the full-load water rate at the steam pressure of the acceptance test as determined from the pressure correction graph (Fig. 252). $\mathbf{W}_S$ = the full-load water rate at the temperature of the superheat of the acceptance test as determined from the superheat correction graph (Fig. 253). $\mathbf{W}_V$ = the full-load water rate at the vacuum of the acceptance test as determined from the vacuum correction graph (Fig. 254). $\mathbf{W}_C$ = the corrected water rate; that is, the water rate after correction from the acceptance-test steam conditions to the steam conditions of the guarantee. $\mathbf{W}_T$ = the water rate as determined by the acceptance test. All water rates are expressed either in pounds per kilowatt hour or per brake horsepower hour.

NOTE.—THE NET CORRECTION FACTOR, C, For. (55), is the algebraic sum (see example below) of the individual correction factors that must be applied to correct for the change in the water rate which will be caused by a change in the steam pressure, superheat, or vacuum. In applying For. (56), it is assumed that the steam consumption at fractional loads will be changed by the same percentage as at full load for the same change in pressure, superheat, and vacuum. This assumption is, for all practical purposes, true within the range of from 50 to 125 per cent. of full-rated load. The method of application of these formulas is explained by the example below.

EXAMPLE.—A 500-kw., 3,600-r.p.m., turbo-generator was sold under the guarantee (Sec. 285) that when operating at rated speed at a steam pressure of 150 lb. per sq. in., gage, 50° F. superheat, and a 28-in. referred vacuum (Sec. 231), it will have the following water rates at the various loads:

CONDITIONS = 150 lb. per sq.-in. gage; 50° F. superheat; 28-in. vacuum.

Load in kw	250.0	375.0	500.0
Per cent. of rated load	50.0	75.0	100.0
Guaranteed water rate in lb. per kw.-hr	20.2	18.3	17.4

When the acceptance test was made—at 50, 75, and 100 per cent. of the rated full load at the rated speed—under a steam pressure of 175 lb. per sq.-in. gage, 100° F. superheat, and a 27-in. referred vacuum, the turbine was found to have the following water rates:

CONDITIONS = 175 lb. per sq.-in. gage; 100° F. superheat; 27-in. vacuum.

Load in kw	250.0	375.0	500.0
Per cent. of rated load	50.0	75.0	100.0
Water rate in lb. per kw.-hr. by acceptance test	18.9	16.9	16.5

The full-load correction graphs (Figs. 252, 253, and 254) for pressure, superheat, and vacuum corrections, are furnished by the turbine manu-

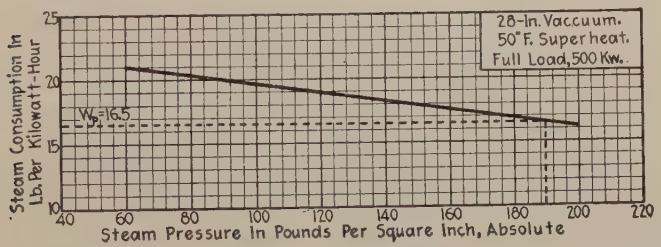


FIG. 252.—Graph for steam-pressure correction of a 500-kw. turbine. This graph shows the performance of the turbine at different supply-steam pressures but with the vacuum constant at 28 in. and with the superheat constant at 50°F.

facturer for the particular turbine which is specified in the guarantee. What is the net correction factor? Correct the water rate for each of the various loads as determined under the steam conditions of the accept-

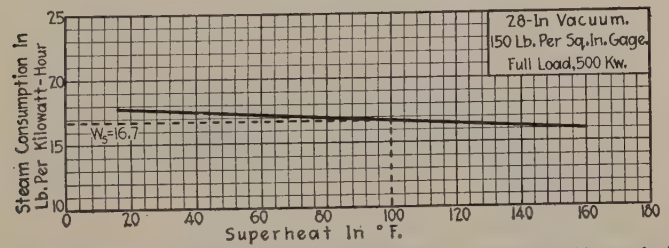


FIG. 253.—Graph for superheat correction of a 500-kw. turbine. This graph shows the performance of the turbine at different superheats but with the supply-steam pressure constant at 150 lb. per sq. in. gage and the vacuum constant at 28 in.

ance test to the conditions of the guarantee specification.

SOLUTION.—The specification guarantees a water rate of 17.4 lb. per kw.-hr. at full load, W_G of For. (55) = 17.4. From Fig. 252, W_P at

175 lb. per sq. in., gage (189.7 lb. per sq. in., abs.) = 16.5 *lb. per kw.-hr.* From Fig. 253, W_S *at* 100° *F. superheat* = 16.7 *lb. per kw.-hr.* From Fig. 254, W_V *at 27-in. vacuum* = 18.5 *lb. per kw.-hr.* Therefore, by substitution in For. (55), *the net correction factor*, C, = $(1 - W_G/W_P) + (1 -$

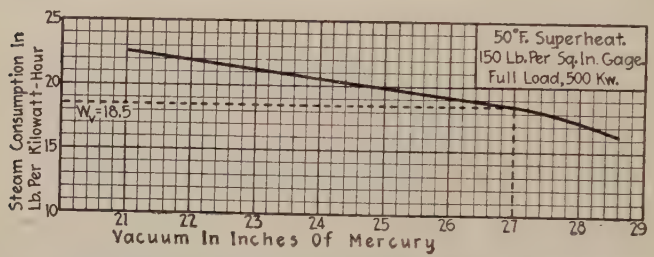


FIG. 254.—Vacuum correction graph for a 500-kw. turbine. This graph shows the performance of the turbine at different vacuums but with the superheat constant at 50° F. and the supply-steam pressure constant at 150 lb. per sq. in. gage.

$W_G/W_S) + (1 - W_G/W_V) = [1 - (17.4 \div 16.5)] + [1 - (17.4 \div 16.7)] + [1 - (17.4 \div 18.5)] = (1 - 1.054) + (1 - 1.042) + (1 - 0.940) = -0.054 - 0.042 + 0.060 = -0.036.$

The water rate, W_T, at full load as determined by the acceptance test is 16.5 *lb. per kw.-hr. The net correction factor as determined above* =

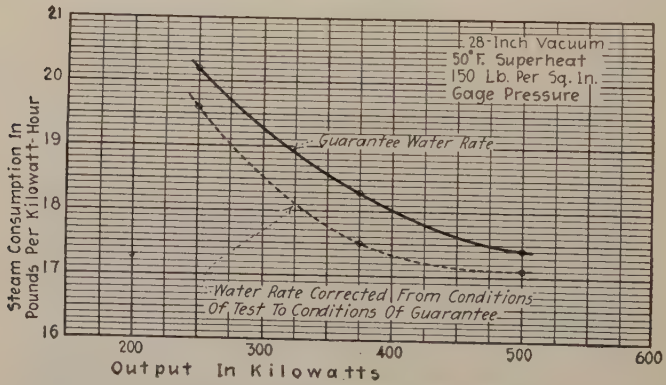


FIG. 255.—Graphs showing: (1) The guaranteed water rate. (2) The water rate as corrected, from the steam conditions obtaining during the test, to the steam conditions on which the guaranteed water rate is based.

-0.036. Therefore, by For. (56), *the corrected water rate at full load*, W_C, = $W_T - CW_T = 16.5 - (-0.036 \times 16.5) = 16.5 + 0.6 = 17.1$ *lb. per kw.-hr.* Similarly, it is found that *the corrected water rate at 75 per cent. full load* = $16.9 - (-0.036 \times 16.9) = 17.5$ *lb. per kw.-hr.* And, *the*

corrected water rate at 50 per cent. full load $= 18.9 - (-0.036 \times 18.9) =$ 19.6 *lb. per kw.-hr.* The following table shows the tabulation of the corrected water rates:

Load in kw	250.0	375.0	500.0
Per cent. of rated load	50.0	75.0	100.0
Corrected water rate in lb. per kw.-hr	19.6	17.5	17.1

By comparing the corrected water rates at the various loads with the guaranteed water rates at the corresponding loads, it is found that the water rates as determined by the acceptance test are lower than those which are guaranteed by the manufacturer. The water rates as determined by test and those which are guaranteed by the manufacturer may be readily compared by plotting a graph of each, against the load in kilowatts or brake horsepower. In Fig. 255, the graphs of the corrected water rates and the guaranteed water rates of this 500-kw. turbine are plotted against the loads in kilowatts.

270. Water-rate Correction Graphs For Changed Pressure, Superheat Or Vacuum Applying To Any High-efficiency, Multi-

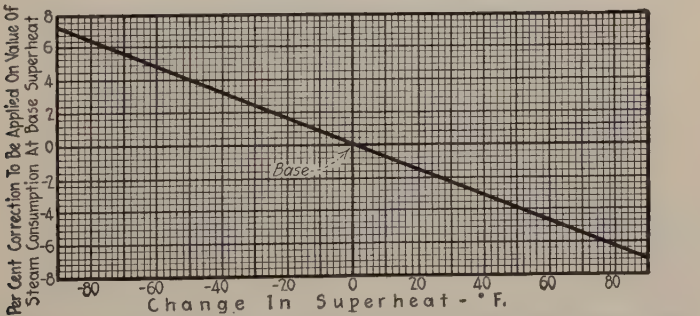


FIG. 256.—Graph for superheat correction for turbine water rates. Supply steam pressure and vacuum are assumed to be constant. "This correction does not apply for superheats below 40° F." (*Allis-Chalmers Mfg. Co.*, June 6, 1922.)

stage, Impulse Or Reaction Turbine are given in Figs. 256, 257 and 258. The accuracy of the results given by them will not be affected by the system of speed regulation which is employed on the turbine. These graphs are used in essentially the same manner as are those of Figs. 252, 253 and 254 except that these are more general in their application.

An Explanation of the graphs of Figs. 256, 257 and 258, as quoted from a letter from the Allis-Chalmers Co. of June 8, 1922 is: "The graphs show the percentage change in steam consumption with changes in the steam conditions of an actual turbine installation. They do not apply if the turbine is altered in a way to render it more suitable for the changed conditions. These correction graphs apply only to the fixed ranges of steam conditions and loads which are, where necessary, specified on them. This matter of limitations is important. It is not believed that it would be feasible to plot a set of usable correction graphs which

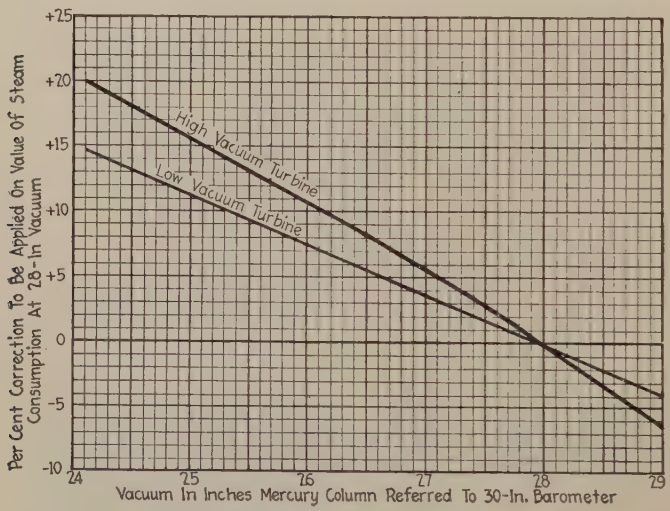


Fig. 257.—Graph for vacuum correction for turbine water rates. Supply-steam, pressure and superheat are assumed to be constant. These corrections apply on most economical load and less only. (*Allis-Chalmers Mfg. Co.*, June 6, 1922.) As defined by the Allis-Chalmers Mfg. Co.: *A high vacuum turbine* is one which, when tested with all conditions constant except vacuum, shows its best Rankine cycle ratio at a vacuum exceeding 27-in. mercury column referred to 30-in. mercury column. *A low vacuum turbine* is similarly one which has its best Rankine cycle ratio at a vacuum below 27 in. mercury column.

would be reasonably accurate for all steam conditions. Such graphs would be too complicated. To insure simplicity the load limitation has not been applied to the vacuum correction graph (Fig. 257). However, it is a fact that the most economical load for a turbine will decrease as the vacuum is decreased and that the vacuum correction for an overload condition will be different from that for a normal load. In applying any correction graphs the most accurate method is to leave the test results without correction and to correct any guarantees or estimates of steam consumption to the steam pressure, superheat and vacuum which prevailed during the tests. The reason for this is that it has become

apparent that the correction of test results to performance guarantee conditions has led to distortion in listing the performances of actual installations."

Example.—Showing the Application Of Pressure Correction Graph Fig. 258. This example is based on information furnished by E. H. Brown of the Allis-Chalmers Company. A 5,000-kw. turbine unit which has its most economical load at 4,500 kw. was sold under a guarantee that when operating at rated speed at a steam pressure of 200 lb.

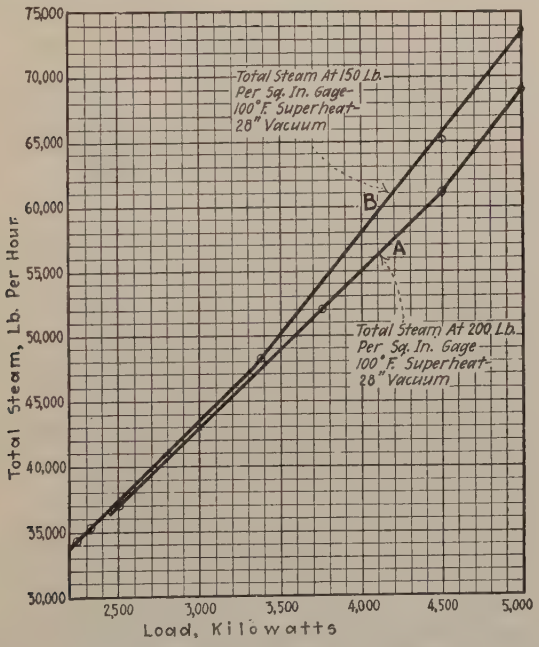


Fig. 257*A*.—Two graphs of the total steam consumptions, in pounds per hour, plotted against the load, in kilowatts, for the turbine unit given in the example of Sec. 270. One graph is for steam at 200 lb. per sq. in., gage, and the other for steam at 150 lb. per sq. in., gage. (From *Allis-Chalmers Co.*, Graph No. St-1,398, September 7, 1922.)

per sq. in., gage, 100° F. superheat, and a 28-in. referred vacuum, it would have the following water rates at the various loads: (If the most economical load for a given turbine is not known, it may be found by plotting the guaranteed water rates against the loads, to which they apply and then finding on this graph the lowest point. This point will correspond to the required—most economical—load.)

Correct the water rate for each of the various loads given in the above table to a steam condition of 150 lb. per sq. in., gage, the superheat and vacuum to be the same as those specified in the guarantee.

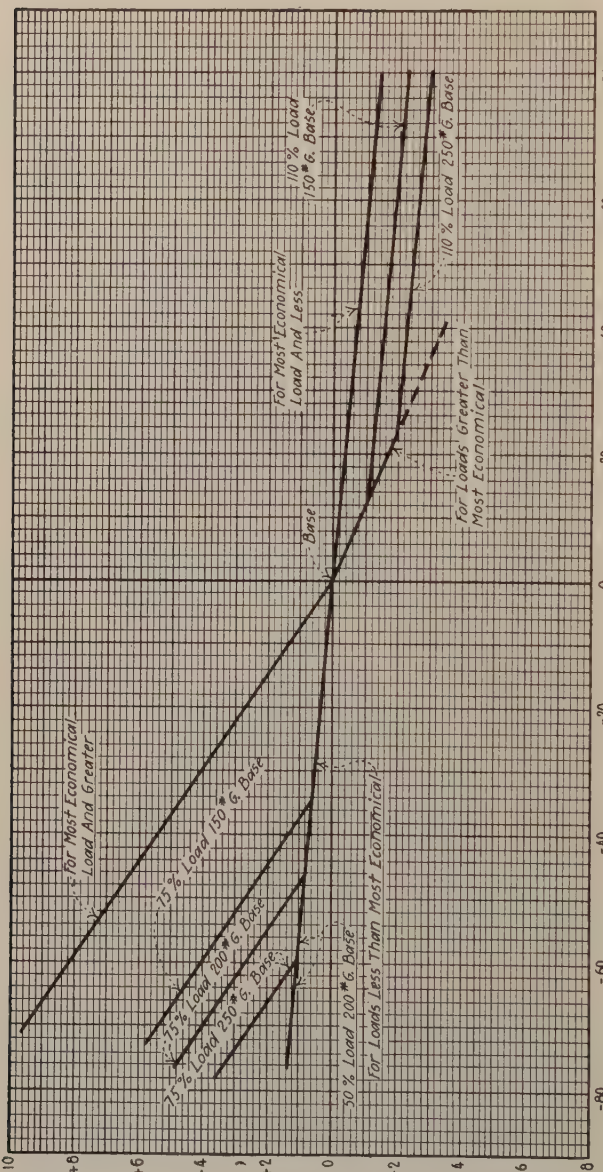


FIG. 258.—Graph for supply-steam-pressure correction for turbine water rates. Superheat and vacuum are assumed to be constant. "These corrections apply over a steam-pressure range of from 150 to 250 lb per sq. in. gage with initial superheats up to 200° F." *The most economical load* refers to the most economical load when the turbine is operating at the base steam pressure. All pressure corrections are made from the base steam pressure. The fractional loads which are indicated are referred to this most economical load at the base pressure. (*Allis-Chalmers Mfg. Co.*, June 6, 1922.)

TABLE 270*A*.—GUARANTEE CONDITIONS 200 lb. per sq. in., gage; 100° F. superheat; 28-in. vacuum.

Load in kw.	2,500	3,750	4,500	5,000
Guaranteed water rate in lb. per kw.-hr	14.8	13.9	13.6	13.8
Total steam—lb. per hr.	37,000	52,100	61,100	69,000

SOLUTION.—Since in this problem the correction is to be made for a change in steam pressure, the correction factors must be taken from the pressure-correction diagram (Fig. 258). But the corrections for the given loads cannot be taken directly from this graph because the loads given in the guarantee are not the certain fractional loads (50, 75 and 100 per cent. and greater) of the most economical load at the base steam pressure which are plotted on the graph. The corrections can be obtained indirectly from the pressure-correction diagram by the following method.

(1) *Plot the guaranteed total steam consumption per hour against the load in kilowatts.*

(2) *From this graph find the guaranteed steam consumption per hour for the certain fractional loads* (50, 75 *and* 100 *per cent. and greater*) *of the most economical load at the base steam pressure, which are plotted on the pressure-correction diagram.*

(3) *Find the guaranteed water rates at the base steam pressures for these fractional loads.*

(4) *Read from Fig.* 258 *the corrections to be applied on the value of the water rate at the base steam pressure to obtain the value of the water rate at the test steam pressure.*

(5) *By means of this correction calculate the water rate and total steam consumption per hour at the test steam pressure.*

(6) *Plot the total steam consumption per hour at test steam pressure against the load in kilowatts.*

(7) *From this latter graph read the total steam consumption per hour at test steam pressure for the given loads.*

(8) *Calculate, from the total steam consumptions per hour obtained in* (7), *the water rates at the test steam pressure for the given loads.*

(9) *Compare the economies of the turbine at the two steam pressures.*

The method outlined above, when applied to the solution of this example will result in the following procedure.

(1) The guaranteed total steam consumptions per hour should be plotted against the given loads in kilowatts. The graph *A*, Fig. 257*A*, will result. Note that the graph consists of two straight lines with different slopes, the charge of slope occurring near the most economical load.

(2) Since the most economical load at the base steam pressure is, in this example, 4,500 kw., the 50, 75, 100 and 111 per cent. loads are 2,250 3,375, 4,500 and 5,000 kw. These will be used as the loads at which corrections will be made. The total steam consumptions per hour at guarantee conditions for these loads can then be read from graph *A*, Fig. 257*A*, as tabulated in Table 270*B*.

(3) The water rates at the base steam pressure for these loads must then be calculated by dividing the total steam consumption in pounds per hour by the loads in kilowatts. These values are given in line 3 of Table 270*B*.

(4) By following up vertically the −50 lb. change-in-steam-pressure line on the pressure-correction diagram (Fig. 258), the correction factors for the four loads may be obtained. Note that the first part of all of the correction curves for loads less than the most economical load coincide along the line marked "*For loads less than the most economical*" and then they branch off from this line, the larger-load curves branching off first. The branching of the 50 per cent. load curve from this line is not shown on Fig. 258 as the diagram is not large enough. The values of these corrections as taken from Fig. 258 are given in Table 270*B*.

(5) The water rates for these loads are then found by multiplying the base water rate by 1 plus the correction factor expressed as decimal. Thus for the load of 2,250 kw., *the water rate at 150 lb. per sq. in., gage, steam pressure* = $15.11 \times [1 + (0.85 \div 100)] = 15.11 \times 1.0085 = 15.23$ *lb. per kw.-hr.*, which checks with the value given in Table 270*B*. From the water rates thus obtained, the total steam consumption per hour at 150 lb. per sq. in. gage pressure can be calculated by multiplying the water rate per kilowatt-hour at each load by the load in kilowatts. The values given in the last line of Table 270*B* will result.

(6) These steam consumptions per hour at 150 lb. per sq. in. gage, steam pressure should then be plotted against their respective loads in kilowatts. The graph *B*, Fig. 257*A*, will result.

(7) The steam consumptions per hour at test conditions for the given loads can then be read from the graph, by following up the vertical line corresponding to the load. The values of these consumptions are tabulated in the Table 270*C*, line 1.

(8) By dividing the total steam consumption in pounds, per hour at 150 lb. per sq. in., gage, steam pressure, for each given load by the load in kilowatts, the water rates in pounds per kilowatt-hour can be obtained. These are listed in line 2 Table 270*C*.

(9) A comparison of the two water rates should be made to show the increase, in per cent., in the water rate. This can be done as follows: The water rate for a load of 2,500 kw. at 150 lb. per sq. in., gage, steam pressure, is 14.92 lb. per kw.-hr. (from Table 270*C*) and that for a steam pressure of 200 lb. per sq. in., gage, was guaranteed as 14.8 lb. per kw.-hr. The change from a pressure of 200 lb. per sq. in., gage, to one of 150 lb. per sq. in., gage, causes, an increase in the water, in per cent. = 100

(14.92 − 14.8) ÷ 14.8 = 12 ÷ 14.8 = 0.85 *per cent.* These values are listed in line 3 of Table 270*C*.

270*B*. TABLE SHOWING VALUES OBTAINED DURING CORRECTION TO CONDITIONS of 150 lb. per sq. in., gage; 100° F. superheat; 28-in. vacuum.

Load in kw.	2,250	3,375	4,500	5,000
Load in per cent. of most economical at base steam pressure.	50	75	100	111
Total steam, lb. per hr. at 200 lb. per sq. in., gage	34,000	47,600	61,600	69,000
Lb. per kw.-hr. at 200 lb. per sq. in., gage	15.11	14.10	13.6	13.8
Correction, in per cent., on 200 lb. per sq. in. gage, steam pressure values (read from Fig. 258 at − 50 lb. change in steam pressure)	+.85	+1.3	+6.77	+6.77
Lb. per kw.-hr. corrected to 150 lb. per sq. in., gage, steam pressure.	15.23	14.29	14.49	14.7
Total steam at 150 lb. per sq. in., gage, steam pressure	34,250	48,200	65,100	73,400

270*C*. TABLE SHOWING COMPARISON of economy at 150 lb. per sq. in. gage, steam pressure, with that at 200 lb. per sq. in gage pressure.

Load in kw.	2,500	3,750	4,500	5,000
Total steam at 150 lb. per sq. in., gage	37,300	53,800	65,100	73,400
Lb. per kw.-hr. at 150 lb. per sq. in., gage	14.92	14.35	14.49	14.7
Per cent. increase in steam consumptions in change of steam pressure from 200 lb. per sq. in., gage, to 150 lb. per sq. in., gage.	0.85	3.2	6.77	6.77

The same method may be used where the steam pressure is increased, and if necessary, correctons for superheat, vacuum and steam pressure may all be applied to one value of economy.

NOTE.—THE "BASE PRESSURE" AND "SUPERHEAT" ARE THOSE FROM WHICH VALUES ARE TO BE CORRECTED.—If the guarantee water rates are to be corrected to test conditions—as is recommended in the preceding

quotation—then the pressure and superheat values which are stated in the guarantee are the "base" values. If, however, the test results are to be corrected to guarantee conditions—as is done in the example under Sec. 269—then the pressure and superheat values which obtained during the test become the "base" values.

QUESTIONS ON DIVISION 13

1. Upon what do the water rates and efficiency of a turbine depend? State the relation in general terms.

2. State the formulas which give the theoretical water rate and thermal efficiency of any turbine.

3. Does the thermal efficiency of a turbine increase in the same proportion as the water decreases when the supply conditions are varied? Give some values to prove this.

4. State what factors determine the theoretical water rate and thermal efficiency of a turbine. What other factor affects the actual water rate and thermal efficiency?

5. State the principal forms in which losses occur in steam turbines. What property of the steam largely affects the amount of the losses?

6. Why are the percentage losses greater in small turbines than in large ones?

7. Are turbines designed for specific steam conditions? How does this fact affect their operation? Explain fully.

8. What would be the action of the steam in a turbine if it were operated under steam conditions much different from those for which it was designed? Explain fully.

9. What effect is produced on the capacity of an existing turbine by increasing its supply steam pressure, superheat, or vacuum?

10. State the approximate factors for calculating the change in water rate due to changes of supply pressure. Superheat. Exhaust pressure.

11. What is the effect on a turbine's efficiency of increasing the supply pressure? Explain fully.

12. What steam pressures are most advisable for turbine operation?

13. How would you compute the effect on a turbine's water rate of changing the supply pressure?

14. What is the effect on a turbine's efficiency of increasing the superheat of its supply steam? Explain why.

15. What superheats are most advisable for turbine operation?

16. What fuel saving may be expected from superheating? Why is very high superheat not economical?

17. How is the most economical superheat for a given plant determined? Draw a typical set of graphs to illustrate the principle.

18. How would you compute the effect on a turbine's water rate of changing the superheat of the supply steam?

19. What effect has the vacuum on the efficiency of a steam turbine? Are there any practical limits?

20. Explain why turbines can more effectively utilize high vacuums than can steam engines. Draw the pressure-volume diagrams for the two classes of machines.

21. What are the usual vacuums that are carried in turbine plants?

22. How is the most economical vacuum for a given plant determined? Draw a typical set of graphs to illustrate the principle.

23. How would you compute the effect on a turbine's water rate of changing the vacuum?

24. What is the effect on a non-condensing turbine's water rate and thermal efficiency of changing the back pressure in the exhaust pipe? How would you compute the effect?

25. What are *performance correction curves?* For what are they used?

26. Explain how you would correct the results of an acceptance test to the conditions of the guarantee? Explain fully.

DIVISION 14

STEAM-TURBINE ECONOMICS AND SELECTION

271. Steam-turbine Economics Is To Be Understood To Mean the study of the *operating costs* (see note below) of steam turbines. The purpose of such studies may be: (1) *To determine the cost of energy,* so that it may be known at what price it may be profitably sold or that the management may know what the energy is costing. (2) *To determine the most desirable turbine for a new plant* or for addition to an existing plant. (3) *To determine whether a turbine is more desirable than a prime mover of some other type.*

NOTE.—THE OPERATING COSTS OF ANY MACHINE are generally grouped into two classes: (1) THE FIXED CHARGES, Sec. 272, which are those expenses that are incidental to the *owning* of the machine; the fixed charges include: (*a*) *Interest* on invested capital. (*b*) *Depreciation.* (*c*) *Taxes and insurance.* (*d*) *Rental and office expense.* (2) THE OPERATING CHARGES, Sec. 273, which are those expenses that arise when the machine is *operated;* they include: (*a*) *Labor and attendance.* (*b*) *Fuel and water.* (*c*) *Repairs and maintenance.* (*d*) *Supplies,* such as waste, oil, and the like. For a more thorough treatment of operating costs, see the author's STEAM-ENGINE PRINCIPLES AND PRACTICE.

272. The Annual Amount Of The Fixed Charges For Turbines varies from about 11 to 15 per cent. of the first cost of the turbine and auxiliaries (installed). The exact percentage to be used in any given case can be determined by taking the sum of: (1) *The current interest rate.* (2) *The depreciation rate,* which is generally assumed as 5 per cent. (3) *The tax rate.* (4) *The insurance rate.* (5) *The rental and office expenses* which are chargeable to the turbine, expressed as percentage of the first cost of the turbine. For the purpose of good bookkeeping, the interest and rental should, rightfully, always be charged against the operation of the turbine whether it is actually paid out or not. In this way only, can the tur-

bine be properly compared with other equipment which is more costly or which occupies a greater amount of space.

NOTE.—THE FIXED CHARGES ARE SO CALLED BECAUSE their amount is the same regardless of whether the machine is in operation or not. In this way they differ, as will be shown, from the operating charges which increase with the output of the machine.

EXAMPLE.—A turbine installation cost $20,000. If money can be borrowed at 6 per cent., the tax rate is 1½ per cent., the insurance rate is ½ per cent., and if the rental and office expenses amount to $400 per year, what is the annual amount of the fixed charges? SOLUTION.—The amount of the rental and office expense is $400 \div 20{,}000 = 0.02 = 2$ *per cent.* Assume that depreciation is 5 per cent. Hence, the *annual fixed charges* $= \$20{,}000 \times (6 + 5 + 1.5 + 0.5 + 2) \div 100 = 20{,}000 \times 0.15 =$ $3,000. Hence it costs the owner of this turbine $3,000 a year merely to own it, whether or not it is operated.

273. The Unit Operating Charges Of Turbines Vary Widely And Depend On Many Things; see following note and Table 274. Reviewing the items (note under Sec. 271) which constitute the operating charges to note how these items may vary, it follows that: (1) The unit labor and attendance expense will vary with the size of the plant and the load which the plant carries because one attendant can generally care for the generating unit whether it has large or small capacity or whether it runs at full or partial load; also, very frequently one attendant can just as easily care for several machines as for only one. (2) The unit fuel and water expense depends upon the efficiency of the boiler, the cost of the coal and the method of handling and firing it, the water rate of the turbine, the quantity of cooling water required if any, the cost of water or the distance it must be pumped. (3) The unit maintenance and repair expense depends on the amount of repairs or maintenance which are necessary and upon the output of the machine. (4) The cost of supplies varies somewhat but, since this item is always small, it is unnecessary to dwell upon it at this point.

NOTE.—THE UNIT CHARGES FOR TURBINES ARE found by dividing the total charges over a certain period of time by the number of energy units which are produced during that period. Unit charges are generally computed on a yearly or monthly basis and on the basis of kilowatt-hours or horsepower-hours produced. The sum of the several unit charges is called the *unit operating cost.*

274. Table Showing Operating Charges For Two Power Plants in a given month as taken from the records of the operating company; station *A* consisted of ten 500-hp. boilers

	Station *A*	Station *B*
Kw.-hr. generated	1,061,000.00	1,210,750.00
Tons of coal	2,775.00	2,437.37
Tons of ash	555.00	322.10
Lb. water evaporated	40,600,000.00	35,359,500.00
Lb. water evaporated per lb. coal	7.32	7.25
Lb. coal per kw.-hr	5.23	4.03
Lb. water per kw.-hr	.....	29.20
Gal. engine oil per 1,000 kw.-hr	3.62	0.59
Gal. cylinder oil per 10,000 kw.-hr	1.74	0.39

Total operating charges, in dollars, and operating charges per kw.-hr., in cents

	Total	Per kw.-hr.	Total	Per kw.-hr.
Superintendence	$122.42	0.014	$250.10	0.020
Repairs:				
Dynamos and appliances	171.33	0.019	10.84	0.001
Engines	.....	.....	.....	.....
Boilers	1,017.48	0.115	299.81	0.024
Pumps, pipes, fittings, and miscellaneous	8.80	0.001	22.15	0.002
Operating boilers	880.92	0.100	392.13	0.033
Operating engines and dynamos	693.66	0.079	390.00	0.032
Supplies	5.47	.....	44.80	0.004
Water	482.21	0.055	99.75	0.008
Lubricants and waste	220.12	0.025	42.50	0.004
Miscellaneous expense	291.24	0.033	60.08	0.005
Total, except fuel	3,893.65	0.441	1,612.16	0.133
Coal	2,635.75	0.298	2,177.44	0.180
Coal labor, car to boiler room	198.62	0.022	114.62	0.009
Total cost	$6,728.02	0.761	$3,904.22	0.322
Average cost of coal on floor of boiler room	$1.0214	.....	$0.94	.....

with hand-fired furnaces, no coal-handling apparatus, burning Illinois screenings, and having 5,000 hp. of reciprocating engines; station *B* was a modern steam-turbine plant with coal- and ash-handling apparatus, economizers, superheaters, and also burning Illinois screenings. (From Gebhardt's STEAM POWER PLANT ENGINEERING.)

275. The Unit Operating Cost For A Turbine Depends On The "Load Factor" (Fig. 259). The load factor is the ratio of the average power delivered by the turbine over a certain

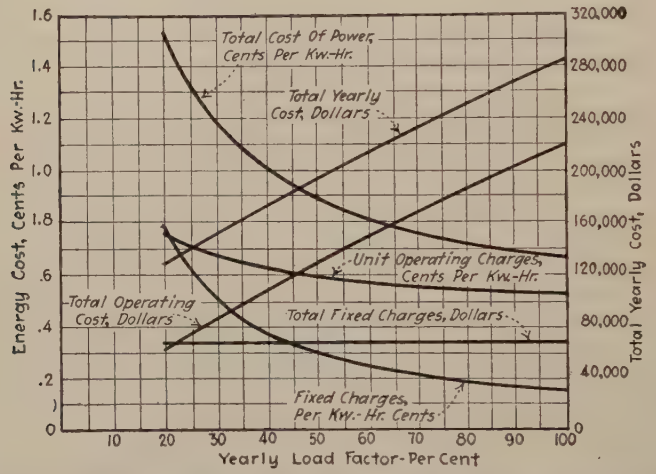


FIG. 259.—Graphs showing how load factor influences the cost of generating energy. Costs at switchboard for a 7,500-kw. steam electric central station. This is from Gebharts, *Steam Power Plant Engineering*.

time period to the maximum power-demand imposed on the turbine during that time period. That is:

$$\text{Load factor} = \frac{\text{Average power}}{\text{Maximum demand}} \quad \text{(decimal)} \tag{57}$$

Load factors are expressed as: (1) *Daily load factors.* (2) *Weekly load factors.* (3) *Yearly load factors.* In Fig. 259, the yearly load factor is used. As is shown by Fig. 259, the total yearly amount of the fixed charges is independent of the load factor whereas the total operating charges increase as the load factor increases, but not directly. Also, the unit fixed charges

and unit operating charges decrease as the load factor is increased. Hence, the unit operating cost varies very widely with different load factors. For a more complete discussion of load factor, demand factor and similar quantities see the author's Central Stations.

Example.—If a plant generates 2,400 kw.-hr. of energy during a 24-hr. period and the maximum demand during that period is 150 kw., what is the load factor for this period? Solution.—*Average power* = *kw.-hr./hr.* = 2,400/24 = 100 *kw.* Hence the *load factor* = *Average power/Maximum demand* = 100/150 = 0.675 or 67.5 *per cent.*

Note.—The Lower The Load Factor, The Greater Will Be The Required Capacity Of The Generating Equipment, For A Given Average Load. If the probable energy required of a plant during a given period is known and the probable load factor is also known, then the probable maximum demand which will be imposed on the generating equipment can be computed thus:

Example.—A plant must generate 500,000 kw.-hr. each month. The probable monthly load factor is 60 per cent. What will be the maximum demand on the plant? In other words what maximum power output must the generating equipment be capable of handling? Solution.—*Maximum demand* = *Average power/Load factor* = 500,000 ÷ (24 × 30)/0.60 = 1,116 *kw.*

276. The Operating Costs Of Turbines Are Generally Computed And Included Together With Those Of The Boilers.—This is done because it would be very difficult, if not impossible, to determine specifically the fuel expense which is properly chargeable to the turbine. Hence, no attempt is generally made to separately determine the costs of the turbine. Instead, the operating cost of the entire plant is generally computed by adding together the boiler-room and turbine-room operating costs. The unit operating cost is then determined for the entire plant. This unit operating cost is then useful for comparison between the turbine plant and a steam-engine plant or an internal-combustion-engine plant.

277. In Selecting A Prime Mover For Any Given Service, consideration must be given to the following factors: (1) *Adaptability;* that is consideration must be given to the distinctive advantages and disadvantages, see Table 287, of the various plants which are being investigated. (2) *Reliability.*

(3) *Economics;* that is, the operating costs (Sec. 271) of the various plants must be studied. In the following sections, the above factors will be discussed principally as they apply to steam-turbine selection. Also, since the selection of a steam turbine generally involves a decision between a steam engine and a turbine, the following discussion will treat principally of the relative merits of these two prime movers.

278. To Render The Steam Turbine Adaptable To Various Services has been the aim of turbine engineers during recent years. Formerly turbines were only designed to run at very high rotative speeds (several thousand revolutions per minute) and hence could be used only with reduction gears to drive relatively high-speed machinery such as electric generators. Today, however, turbines are designed for rotor speeds as low as 1,200 r.p.m. and, with reduction gears, are being used to drive even the slowest-speed machinery. Inherently, however, the turbine is best adapted for driving high-speed machinery which must operate at a constant rotative speed. Hence, its most extensive use is for driving electric generators, centrifugal pumps, blowers, and like high-speed machinery. Furthermore, as has been shown in Div. 9, the turbine is adapted for almost any steam pressures and can be operated condensing or to exhaust against back pressures.

Note.—The Steam Turbine Is Not Reversible And Cannot Be Efficiently Operated At Variable Speeds.—These two limitations are practically the only ones which need ever rule out the turbine from the viewpoint of adaptability. However, even these have been somewhat overcome in marine practice where, for reversing, a separate turbine is employed and, to secure maximum efficiency, full speed is maintained whenever possible.

279. Modern Turbines Are Very Reliable.—Because of the small number of bearing surfaces in a turbine and because of its purely rotational motion, the lubrication of the bearings can be made very positive, Div. 10, and the wear is inappreciable. If kept in proper alignment and carefully operated, a steam turbine is more reliable than a prime mover of any other kind.

280. Table Of Approximate Water Rates And Unit Costs Of Turbo-generators.—This tabulation is for turbines driving 60-cycle, 2,300-volt, alternating-current generators. The speeds are: 50 to 3,500 kw., inc., 3,600 r.p.m.; 4,000 to 20,000 kw., inc., 1,800 r.p.m.; 25,000 kw. and over, 1,200 r.p.m. (Water rates from Marks' MECHANICAL ENGINEERS' HANDBOOK. Prices from Fig. 266.)

Condensing						Non-condensing		
Steam at 150 lb. per sq. in. gage, 28-in. vacuum			Steam at 200 lb. per sq. in. gage, 150° F. superheat, 29-in. vacuum			Steam at 150 lb. gage, atmospheric exhaust		
Rating, kw.	Water rate, lb. per kw.-hr. including excitation	Price per kw., dollars	Rating, kw.	Water rate, lb. per kw.-hr. including excitation	Price per kw., dollars	Rating, kw.	Water rate, lb. per kw.-hr.	Price per kw., dollars
50	32.0 to 42.0	....	4,000	12.0 to 13.3	16.5	50	52.0 to 60	70.0
100	21.2 to 27.5	51.0	5,000	11.9 to 13.0	16.0	100	43.5 to 50	56.0
200	19.5 to 25.0	39.0	7,500	11.6 to 12.6	16.0	200	38.4 to 44	44.0
300	18.7 to 22.5	33.0	10,000	11.2 to 12.3	15.5	300	36.5 to 42	37.0
400	18.2 to 21.0	29.0	12,500	11.1 to 12.0	15.3	400	36.2 to 40	32.0
500	17.7 to 19.3	27.0	15,000	11.1 to 11.8	15.0	500	34.1 to 38	30.0
600	17.4 to 19.0	25.0	17,500	11.0 to 11.6	14.8	600	33.4 to 37	28.0
750	17.0 to 18.8	23.0	20,000	11.0 to 11.4	14.5	750	32.7 to 36	25.5
1,000	16.5 to 18.6	22.0	25,000	10.9 to 11.3	14.3	1,000	31.7 to 35	23.0
1,250	16.2 to 18.3	20.5	30,000	10.8 to 11.2	14.0	1,250	31.0 to 34	22.0
1,500	16.0 to 18.0	19.0	35,000	10.7 to 11.1	....	1,500	30.7 to 33	21.0
2,000	15.7 to 17.7	18.0	40,000	10.6 to 11.1	....	2,000	30.1 to 32	20.5
2,500	15.5 to 17.4	17.5	50,000	10.5 to 11.0	....	2,500	29.4 to 31	....
3,000	15.4 to 17.1	17.0	60,000	10.5 to 10.9	....	3,000	28.9 to 30	....
3,500	15.3 to 16.9	16.5	75,000	10.4 to 10.8	....	3,500	28.6 to 29	....

281. The Efficiency Or Steam Economy Of A Turbine Depends Principally On Its Size And Steam Conditions.—The effect of size is partially illustrated in Figs. 264 and 265; turbines of larger capacity than those represented in these graphs show even better efficiencies—the very large condensing turbines have water rates of about 11 lb. of steam per kw.-hr.; see Table 280. The effects of steam conditions—pressure, vacuum, and superheat—have been discussed in Div. 13. There seems to be little difference, if any, between the efficiencies of impulse and reaction turbines of equal capacity; reaction turbines, however, are not practicable in sizes smaller than about 125 kw.

282. The Efficiency Or Steam Economy Of Turbines At Fractional Loads (Fig. 260) is very much better than that of engines. Figure 260 shows that the steam rate increases more as the load is decreased with small turbines than with large ones. The high efficiency of turbines at light loads is particularly advantageous in electric power stations where turbines must frequently be operated at fractional loads so as to be ready for a sudden increase in station load.

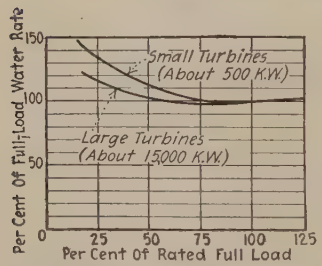


Fig. 260.—Graphs showing approximate variation of the steam consumption of turbines with variations of load.

Note.—The Capacity Rating Of A Turbine Generally Means Very Little.—Turbines often are most efficient at loads which are considerably less than their rated capacity and are usually capable of supplying considerably more power than their rating. Large turbines are often rated at the maximum load which their generators are capable of developing *continuously* (see Fig. 273). But this, too, is not always the basis of the rating. Hence, the meaning of a turbine rating is often quite indefinite.

283. The Economy Of Low- And Mixed-pressure Turbines is, generally speaking, better than that of turbines of any other class. This is because that, with these turbines, the capacity of an existing plant can be greatly increased without increasing the fuel consumption. By utilizing the exhaust

steam from non-condensing engines, pumps, or other equipment, the capacity of a plant can often be increased by 80 to 100 per cent. without any increase of the boiler capacity. Where condensing engines are in use, these may be run non-condensing and their exhaust then utilized in a turbine—an increase in capacity of 40 to 50 per cent. may thus be obtained with but a slightly greater amount of steam consumed. See Div. 9 on low- and mixed-pressure turbines.

Note.—The Use Of Separate High-pressure Non-condensing And Low-pressure Turbines Is Not Advisable; the very-large capacity

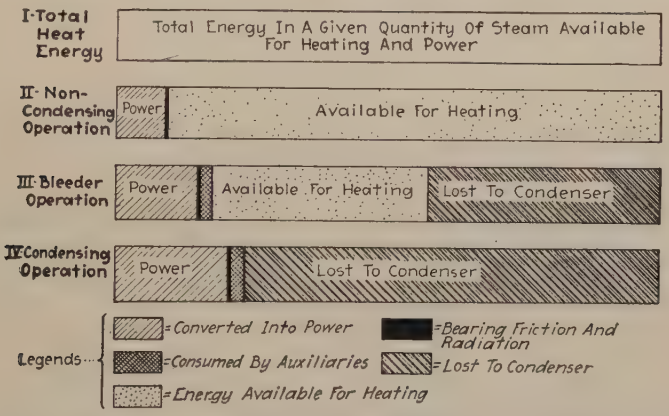


Fig. 261.—Chart showing approximately the disposition of the heat energy in a given quantity of steam when it is used in turbines of different types. The bleeder turbine operation (*III*), can, on a moment's notice, be changed to either that of the non-condensing (*IV*) turbine or any condition intermediate between *II* and *IV*—as power and heat requirements may demand.

compound units, Sec. 68, are considered as being single units. Such an arrangement, although efficient in its use of steam, is not commercially economical because it necessitates a duplication of turbine and generator units—it is usually found that one high-pressure condensing turbine is better. Exhaust-steam turbines should, therefore, only be employed where profitable use can be made of the exhaust steam from *existing* steam-using equipment.

284. The Economy Of Bleeder Turbines, Fig. 261, (see also Div. 9) lies in the fact that, by them, low-pressure steam is made available for heating or industrial services after the steam has been first used very efficiently to generate electrical energy in the bleeder turbine unit. By so arranging the load

on a bleeder turbine that the turbine always consumes (receives) considerably more steam than is extracted from it the turbine can thereby be made more efficient in its use of steam than would be a non-condensing turbine which consumed only the amount of steam that is necessary for low-pressure heating or the like. Bleeder turbines are, therefore, being used more and more as house turbines in large power stations—the auxiliaries being driven largely by electric motors which are supplied with energy from the generator which the bleeder turbine drives.

Note.—For The Most Economical Application Of A Bleeder Turbine in an electrical generating station, it should be operated in conjunction with another (condensing) turbine. The total load is divided between the two units. The load on the bleeder turbine can then be changed, from time to time, as is necessary to insure that this turbine will always "bleed" sufficient low-pressure steam to satisfy feed-water or other heating requirements.

285. To Predict The Steam Rate Of A Contemplated Turbine, the method of Sec. 15 may be used for the first estimate; or it may be read from Table 280. The exact water rate, however, can best be determined by applying to various manufacturers for their *guarantees*. Manufacturers generally specify steam economies which their turbines will actually exceed by a slight amount. This they do to be on the safe side. Having the builders' guarantees one may then make his final calculations. When bleeder or mixed-pressure turbines are contemplated, their low-pressure steam rates must very often be estimated; hence, undue accuracy in their water-rate calculations should be avoided.

286. The Relative Economies Of Steam Turbines And Steam Engines depend, to a great extent, upon local conditions. Because they generally operate under different conditions it is often difficult to make reasonable comparisons between the two. Certain items of economy, however, are quite general in that they hold for nearly all comparisons—these items have been included in Table 287. Since the first cost of turbines is less than that of engines of equal capacity, the interest, taxes, insurance and depreciation charges are correspondingly less. The rental charges are also less, because of the fact that the

turbine occupies less space; see Figs. 262 and 263. Likewise with the other economy items given in Table 287. Practically the only item of economy which is not given in Table 287 is

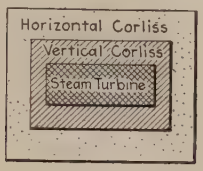


FIG. 262.—Comparative floor space occupied by steam engines and turbines.

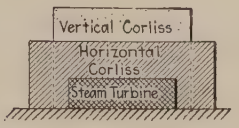


FIG. 263.—Comparative head room necessary for steam engines and turbines.

that of operating efficiency or steam economy; this item is treated in Secs. 288 and 289.

287. Table Of Advantages And Disadvantages Of Steam Turbines And Steam Engines.

STEAM TURBINE	STEAM ENGINE
ADVANTAGES	DISADVANTAGES
Low first cost.	Greater first cost.
Low maintenance and attendance.	Greater maintenance and attendance.
Economy of space and foundation.	Requires more space and larger foundation.
Clean exhaust steam.	Oil in exhaust steam.
No vibration due to reciprocating parts.	Reciprocating parts cause vibration.
Uniform angular velocity.	Angular velocity varies during each stroke. Heavy flywheel required.
High efficiencies for large variations in load.	Decreased efficiency at fractional loads.
Can utilize steam at high temperatures.	High temperatures give trouble.
DISADVANTAGES	ADVANTAGES
Cannot be made reversible.	Can be made reversible.
Speed too high for many services.	Runs at low angular speed.
Runs at constant speed.	Can be run at variable speeds.
Condenser requires much water.	Condenser requires less water.

288. The Relative Steam Economics Of Non-condensing Turbines And Engines are illustrated in Fig. 264 for full-load operation; see also Table 280. It is well to note that the non-condensing turbine is not as efficient as the non-condensing engine. However, at fractional loads (Sec. 282), the turbine's efficiency is more nearly equal to the engine's. As is shown by Fig. 264, the efficiency of the turbine in the larger sizes is also more nearly equal to that of the engine than in the smaller

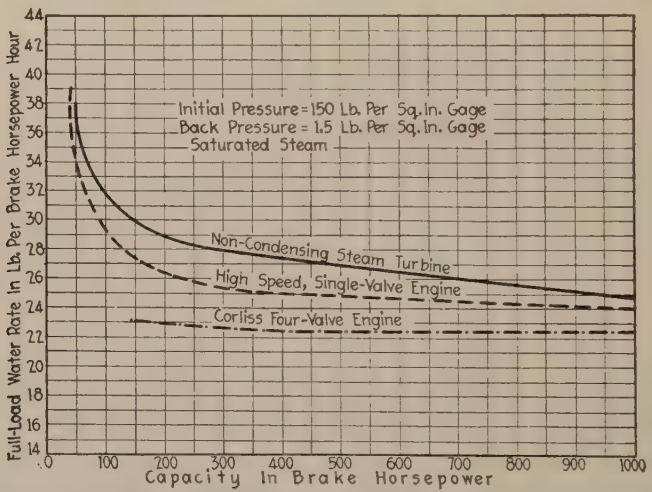


Fig. 264.—Graphs showing comparative steam economies at full load of average non-condensing steam engines and steam turbines. These graphs show the water rates, in pounds per brake horsepower-hour, for engines and turbines supplied with dry saturated steam at 150 lb. per sq. in gage and exhausting against a back pressure of 1.5 lb. per sq. in. gage.

sizes. Furthermore, the turbine is more efficient with high steam pressures whereas the steam engine is more efficient with lower steam pressures. Although, as shown above, the efficiency of the non-condensing engine exceeds that of the non-condensing turbine, this is not to be taken to mean that the overall economies are so related. Because of the turbine's lesser first cost, attendance, and maintenance expense, and because of its other advantages (Table 287), the turbine is, in many cases, more economical than the more efficient steam engine.

288A. Turbine Steam Rates Are Also Less Likely To Increase With Years Of Service Than Are Engine Steam Rates.—This is because the only wearing parts of the turbine are the bearings, nozzles, and blading. The nozzles and blading do not "fit tight," even when the turbine is new. A small amount of wear, due to steam erosion, of these nozzles and blades will not produce excessive steam leakage as will a small amount of wear on engine valves or cylinders.

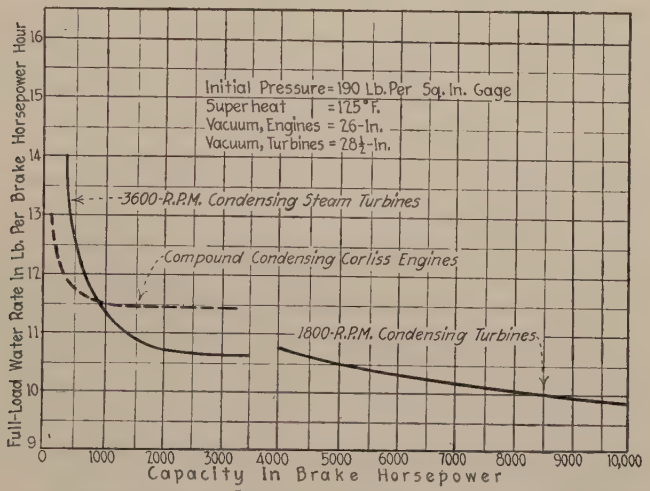


FIG. 265.—Graphs showing comparative steam economies at full load of condensing engines and turbines. These graphs are based on steam supplied at 190 lb. per sq. in. gage and 125° F. superheat and on a vacuum for the engines of 26 in. and for the turbines 28.5 in. The condensing steam turbine is more efficient than the compound condensing steam engine in capacities of 1,000 hp. and larger.

289. The Relative Steam Economies Of Condensing Turbines And Engines, at full load, are illustrated in Fig. 265; see also Table 280. The efficiencies of the engines are seen to be better in the capacities below 1,000 hp., whereas in larger capacities the turbines show the better efficiencies. This comparison, it should be noted, is made with a greater vacuum on the turbines than on the engines; this is done because the turbine is most economical at greater vacuums than is the steam engine; Sec. 264. *The turbine*, therefore, *requires more cooling water than does the engine* and is less desirable where only a limited supply of cooling water is available or where the circulating water must be recooled in ponds or towers.

290. Table Showing Applicability Of Steam Turbines And Engines In Units Of Small Capacity.—This table is based on a paper by J. S. Barstow before the A.S.M.E. in Dec., 1915 and applies chiefly to units of 500-hp. or less capacity.

Turbines	Engines
1. Condensing Units, Direct-connected For: (*a*) 60-*cycle generators* in all sizes. (*b*) 25-*cycle generators* above 1,000-kw. capacity. (*c*) *Centrifugal pumping machinery* operating under substantially constant head and quantity conditions and at moderately high head, say from 100 ft. up, depending on the size of the unit. (*d*) *Fans and blowers* for delivering air at pressures from 1½-in. water column to 30 lb. per sq. in.	1. Non-condensing Units, Direct-connected Or Belted For: (*a*) *Electric generators* of all classes, excepting exciter sets of small capacity unless belted from the main engine. (*b*) *Centrifugal pumping machinery* operating under variable head and quantity conditions and at relatively low heads, say up to 100 ft., depending on the capacity of the unit. (*c*) *Pumps and compressors* for delivering water or gases in relatively small quantities and at relatively high pressures—in the case of pumps at pressures above 100 lb. per sq. in., compressors above 1 lb. per sq. in. (*d*) *Fans and blowers* (including induced-draft fans) for handling air in variable quantities and at relatively low pressures, say not over 5-in. water column. (*e*) *Line shafts of mills*, where the driven apparatus is closely grouped and the load factor is good. (*f*) *All apparatus requiring reversal* in direction of rotation, as in hoisting engines, and the like.
2. Non-condensing Units, Direct-connected For All The Above, Purposes, In Those Cases Where: (*a*) *Steam economy* is not the prime factor or the exhaust steam can be completely utilized. (*b*) *Oil-free exhaust steam* is desirable or essential.	2. Condensing Units, Direct-connected or belted, for all the above purposes, particularly where: (*a*) *The condensing water* supply is limited. (*b*) *The water must be recooled* and recirculated.
3. Geared Units, Either Condensing or Non-condensing, for all the above applications and, in addition, many others which would otherwise fall in the category of the steam engine, on account of the relatively slow speed of the apparatus to be driven.	

291. The First Costs Of Steam Turbines of different capacities are given approximately in Table 280 and Fig. 266. The values given here must be understood to be only indications and subject to the influence of local conditions and market fluctuations as it is impossible to give prices which will be even nearly correct for any length of time due to the rapid change of prices. These prices are not intended to be accurate at any future date but, they may, however, be used for preliminary estimates of power-plant cost as they show how the price varies with the size of the unit. This relationship remains practically the same regardless of the change in price. If at any future date the percentage change of the average

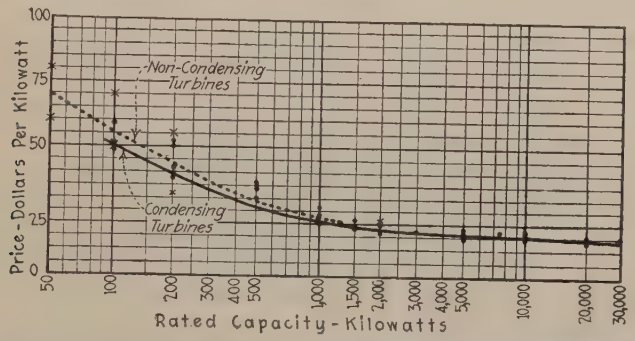


FIG. 266.—Showing approximate prices of turbo-generator units of different capacities. (The dots represent prices quoted on condensing units. The crosses represent prices quoted on non-condensing units. Prices are as of spring, 1922. Some of the units are not equipped with direct-connected exciters, but in most cases the price includes exciter. The omission or addition of the exciter makes little difference in the price per kw. Condensers are not included.)

price based on that given here (spring 1922) is known, the price of any unit at that date can be approximately found by multiplying the price given here by that percentage and making this correction to the price here given. Whenever reasonably accurate prices are required they should be obtained from the manufacturers. The graph of Fig. 266 shows remarkably well how the cost per kilowatt decreases with increased size of the unit. It will be noted from Fig. 266 that the price per kilowatt decreases very rapidly with an increase in the size of the unit for units between 50 and about 1,000 kw. capacity. Above 1,000 kw. capacity, the price per kilowatt does not

decrease very much for an increase in capacity. Above 30,000 kw. capacity the price per kilowatt is practically constant.

292. The Steps To Be Taken In Selecting A Prime Mover For A Given Service are: (1) *Determine the load factor,* Sec. 275, *and the hourly load variation* if possible; if it is not possible to accurately determine the load variation, then try to obtain the probable load variation from some similar plant. (2) *Determine the maximum load;* in new plants, the maximum load must often be estimated. (3) *Select the most desirable capacities* of units; this should be done with a view toward always operating each unit at its most economical load. Generally speaking, the fewer units in a plant the better, provided always that there is sufficient generating capacity to carry the maximum peak with the largest unit out of service. (4) *Get costs and performance guarantees* (Sec. 294) for the different units of each type which is being considered; this usually requires the making of tentative building and machinery layout drawings of the arrangements which are under consideration. (5) *Calculate the unit operating costs for each type* over a yearly period; to do this, estimates of the operating charges must be made. (6) *Tabulate the estimates and decide on the type of equipment which shows the smallest unit operating cost,* or is otherwise most desirable.

The method of selecting a prime mover is explained by the following illustrative example, which is taken from the *National Electric Light Association* Prime Movers Committee's Report for 1921.

Note.—The Values In The Following Example, As It Is Here Used, Are Intended To Illustrate A Method Of Procedure rather than to "prove in" or "prove out" any certain type or class of power-generating equipment. Obviously, the values of the different elements which comprise the total cost will vary in different localities. The costs shown are for the vicinity of New York City in the year 1921. It is only by thus preparing an accurate tabular comparison of the costs of energy, as developed by different types of equipment and under different conditions, that the most economical equipment and steam conditions for a given location can be determined. In the N.E.L.A. report, above referred to, an energy-cost comparative analysis is also given for 200-kw. plants which operate at load factors of 25 and 75 per cent.

EXAMPLE.—It is desired to select the most economical equipment for a generating station which is to furnish electrical energy at the average power-output rates stated in Table I below. The following equipment is to be considered: (*a*) Uniflow engines, (*b*) high-speed counterflow engines, (*c*) turbines, (*d*) Corliss engines, (*e*) Diesel oil engines and (*f*) semi-Diesel oil engines. The most adaptable steam pressures may be assumed as 175 lb. per sq. in. for all units except the Corliss engines for which 150 lb. is to be used. Costs are to be determined for non-condensing (atmospheric exhaust) and for condensing operation both with saturated steam and with steam of 100° F. superheat. The condensing engines are to operate with 26-in. vacuum; the turbines with 28-in. vacuum.

The cost of coal is to be taken at $7.00 per ton, delivered. The heating value of the coal is 13,500 B.t.u. per lb. The oil engines are to be supplied with an oil of 18,500 B.t.u. per lb. heating value which will cost about $3.00 per bbl., delivered. The maximum peak load, assumed to occur only occasionally, is 200 kw. The average 24-hr. daily demand is assumed to vary as follows:

TABLE I.—THE LOADS AND THEIR DURATION

1. Load, in kw.	2. Duration of load, hours	3. Kw.-hr. of energy generated
200	Peak load	.....
180	1	180
140	8	1,120
100	5	500
75	4	300
60	4	240
30	2	60
Totals.....	24	2,400

The plant is assumed to be located in a small town near an adequate supply of water of the proper quality for condenser or oil-engine cooling. Also, a railroad siding is adjacent for the delivery of coal or oil.

SOLUTION.—Proceeding as suggested in Sec. 292, the steps are as follows:

1. DETERMINE THE LOAD FACTOR AND LOAD VARIATION.—The load variation has been determined by comparison with similar plants and found to be as shown in Table I above. The load factor is found thus:

$$\textit{The average load} = \frac{\textit{Total kilowatt-hours generated}}{\textit{Hours duration}} = \frac{2{,}400}{24} = 100 \textit{ kw.}$$

Now, the load factor for this average load will, from For. (57), be:

$$\textit{Load factor} = \frac{\textit{Average load}}{\textit{Maximum demand}} = \frac{100}{200} = 0.50 \text{ or } 50 \textit{ per cent.}$$

2. Determine The Maximum Load.—The maximum load is given in the problem as 200 kw.

3. Select The Most Desirable Capacities Of Units.—To provide

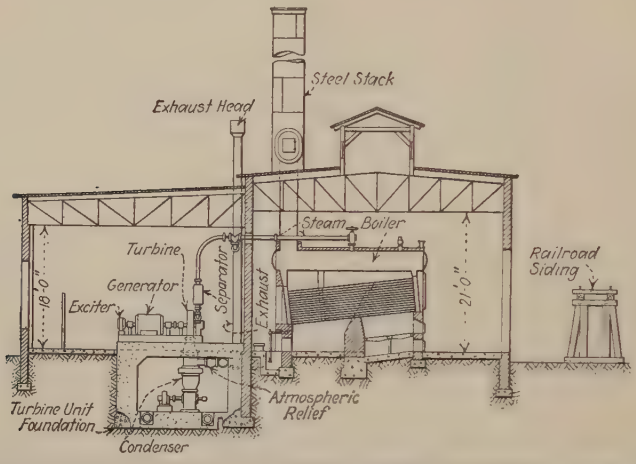


Fig. 267.—Sectional elevation of the 300-kw. and also of the 400-kw. (total capacity) steam turbine generating stations the plan views of which are shown in following illustrations. (N.E.L.A., 1921, *Prime Movers Report.*)

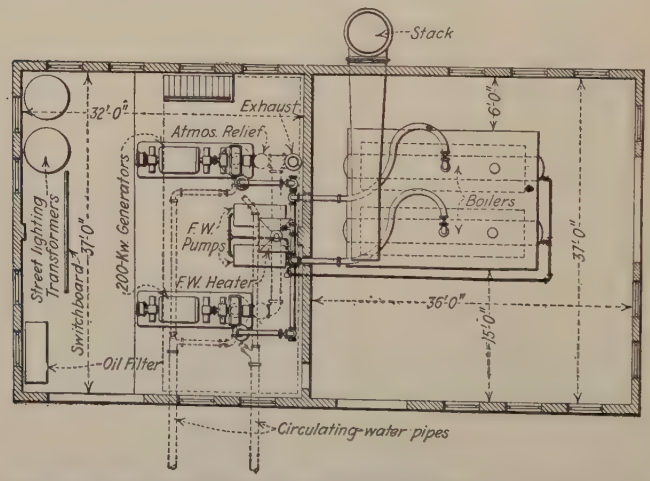


Fig. 268.—Plan view of the 400-kw., total capacity (2–200 kw. units) steam turbine generating station. See preceding illustration for section.

sufficient generating capacity with the largest unit out of service and yet to have only a small number of units, it is thought advisable to con-

sider and make calculations for (*a*) two 200-kw. and (*b*) three 100-kw. generating units of each type.

4. Get Costs And Performance Guarantees For The Different Units.—The building in all cases is assumed to be of brick construction.

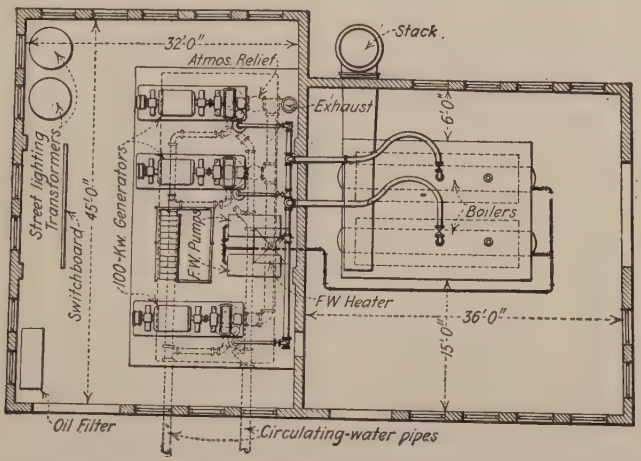


Fig. 269.—Plan view of the 300-kw., total capacity (3–100 kw. units), steam turbine generating station. See preceding illustration, Fig. 267, for section.

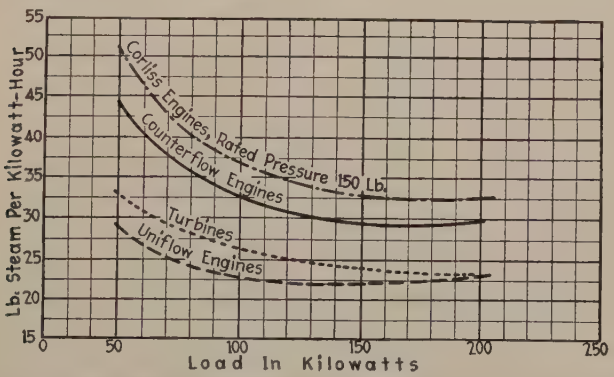


Fig. 270.—Average steam consumptions per kilowatt-hour for 200-kw. condensing steam units.

In all cases except for the belted Corliss engines, the roof trusses are of steel. The station to house the belted Corliss engines is designed with wooden roof trusses and a central line of posts on account of the long span required.

The layouts of the buildings and principal equipment for the turbine plants are given in Figs. 267, 268, and 269. In the N.E.L.A. report

from which this example is taken, layouts are shown for all of the different plants which are considered. From such layouts, contractors can make estimates. The investment costs are tabulated in Table IV below. The steam equipment is found to require boiler capacities as follows: (*a*) For the non-condensing turbines and for the Corliss engines (both condensing and non-condensing), two 200-hp. boilers. (*b*) For all other cases, two 150-hp. boilers. Proposals and performance specifications were obtained from 65 manufacturers and averaged by classes. The average steam (or oil) consumptions were plotted into curves of which Fig. 270 is typical.

5. Calculate The Unit Operating Cost For Each Type.—To do this, the yearly operating costs are first found and later, from these, the unit operating costs are found. The annual fixed charge is assumed to be 15 per cent. of the total investment cost for all plants, this figure including interest, taxes, depreciation and both liability and fire insurance. The fuel costs are thus determined: (*a*) *For the oil-engine plants.* The oil consumed per 24-hr. day was computed by multiplying each item of column 3, Table I, by the fuel rate at the load shown in column 1. (The fuel rate is read from the guarantee curve.) From the daily oil consumption, the annual fuel-oil cost can readily be obtained. (*b*) *For the steam plants.* The steam consumed per 24-hr. day was computed by multiplying each item in column 3 of Table I by the steam rate at the load shown in column 1. Thus, for the 200-kw. condensing turbine plant:

Table II.—Steam Consumption, 200-kw. Condensing Turbine Plant

Load, in kw., from Table I	Water rate, lb. per kw.-hr., from Fig. 270	Kw.-hr. generated, from Table I	Steam consumed, lb.
180	23.5	180	4,230
140	24.5	1,120	27,450
100	26.5	500	13,250
75	29.0	300	8,700
60	31.0	240	7,440
30	38.0	60	2,280
Totals.....	....	2,400	63,350

Now, allowing 9,600 lb. per day for losses due to pipe radiation, drips, and like, 3,200 lb. per day for the boiler-feed pumps, and 9,900 lb. per day for the condenser air and circulating pumps, the *total daily steam consumption* = 63,350 + 9,600 + 3,200 + 9,900 = 86,050 *lb.*

Since, in this example, the load factor is 50 per cent., the boiler efficiency will be about 64 per cent. Also, from steam tables, the total

heat of dry saturated steam at 175 lb. per sq. in. gage is 1,198 *B.t.u. per lb.* If a feed-water temperature of 200° F. is assumed, the heat of the liquid (from steam table) is 168 *B.t.u. per lb.* Hence, *the B.t.u. absorbed per pound of steam* = 1,198 − 168 = 1,030 *B.t.u.* Therefore, with coal of 13,500 B.t.u. per lb. heating value, and a boiler efficiency of 64 per cent., the *evaporation* = 0.64 × 13,500 ÷ 1,030 = 8.39 *lb. steam per lb. of coal.* Therefore, the *daily coal consumption* = 86,050 ÷ (8.39 × 2,000) = 5.13 *tons per day.* Hence, at $7.00 per ton, the *annual coal cost* = 5.13 × 365 × $7.00 = $3,100. (See Table IV.)

The annual labor cost is computed by assuming the required attendants and their probable salaries, thus:

TABLE III.—ATTENDANTS REQUIRED AND SALARIES

Class of employee	Number required		Salary, each, per month
	Steam plant	Oil-engine plant	
Chief engineer	1	1	175
Watch engineers	2	2	125
Oilers	..	3	110
Firemen	3	..	110

Thus, for the 200-kw. condensing plant the *annual cost of labor and superintendence* = 12 × [175 + (2 × 125) + (3 × 110)] = $9,060. Now, since the regular power-plant force of attendants can, ordinarily, attend to the making of repairs about the plant, $1,000 of the annual salaries may be charged to repairs leaving *the annual charge for labor and superintendence* = $9,060 − $1,000 = $8,060; see Table IV.

The annual costs of lubricants, miscellaneous supplies, and pumping cooling water (for the oil engines) were estimated; the estimated values are given in Table IV. The annual costs of repairs were figured at 4 per cent. of the investment costs of the engine plants and at 3 per cent. in the case of turbine plants. Thus, for the 200-kw. condensing-turbine plants the annual repair cost was figured as 3 per cent. of the investment cost for the generating units, condensing equipment, boilers, and feed pumps. The investment cost for this equipment = $19,460 + 7,000 + 13,000 + 1,500 = $40,960. Therefore, *the annual repair cost* = 0.03 × $40,960 = $1,229; see Table IV.

Thus, the *total annual cost of operation* is the sum of annual fixed charges, fuel, labor and superintendence, lubricant, miscellaneous supplies, and repair costs. This sum gives a value of $38,228 as shown in Table IV.

Therefore, the unit operating costs may be computed by the formula:

$$\text{(58)} \qquad \textit{Unit operating or energy cost} = \frac{\textit{Annual operating cost}}{\textit{Energy units delivered per year}}$$

TABLE IV.—SHOWING INVESTMENT AND OPERATING Load Factor 50 Per Cent. Steam Pressure 175 Lb. per Sq. In. Gage for All Engines. Cost of 13,500 B.T.U. Coal $7.00 Per Net

Items	Non-condensing steam prime movers Back-pressure atmospheric							
	Uniflow engine		Counter-flow engines		Turbines		Corliss engine belted	
	2-200 kw.	3-100 kw.	2-200 kw.	3-100 kw.	2-200 kw.	3-100 kw.	2-200 kw.	3-100 kw.
	A—STEAM EQUIPMENT DESIGNED							
INVESTMENT:								
Real estate	$ 2,500	$ 2,500	$ 2,500	$ 2,500	$ 2,500	$ 2,500	$ 2,500	$ 2,500
Brick building	35,200	35,800	35,200	35,800	26,000	28,000	43,000	45,000
Generating units, delivered and erected	31,954	32,250	25,504	26,634	16,750	19,449	23,380	24,450
Switchboards and street lighting transformers	6,500	7,000	6,500	7,000	6,500	7,000	6,500	7,000
Electric wiring and ducts	3,500	4,000	3,500	4,000	3,500	4,000	3,500	4,000
Piping complete	6,500	6,500	6,500	6,500	6,500	6,500	6,500	6,500
Condensing equipment	.....	.....	.....	.....	.....	.....	.....	.....
Foundations, exclusive of building	4,200	3,900	4,200	3,900	2,500	2,800	4,500	4,200
Oil filters and tanks	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800
Railroad siding	3,200	3,200	3,200	3,200	3,200	3,200	3,200	3,200
Boilers, delivered and bricked in	13,000	13,000	13,000	13,000	15,400	15,400	15,400	15,400
Feed water heater	900	900	900	900	900	900	900	900
Feed pumps	1,500	1,500	1,500	1,500	1,500	1,500	1,500	1,500
Steel stack and flues	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800
Motor-driven pump for cooling water	.....	.....	.....	.....	.....	.....	.....	.....
10,000 gal. fuel oil storage tank	.....	.....	.....	.....	.....	.....	.....	.....
Air compressor and tanks	.....	.....	.....	.....	.....	.....	.....	.....
Intake for circulating water	.....	.....	.....	.....	.....	.....	.....	.....
Total investment	$112,554	$114,150	$106,104	$108,534	$ 88,850	$ 94,849	$114,480	$118,250
COST OF OPERATION:								
Fixed charges 15 per cent. on investment	16,883	17,123	15,915	16,280	13,328	14,227	17,172	17,737
Fuel	13,310	13,302	16,290	15,692	20,974	20,902	18,100	17,902
Labor and superintendent	8,060	8,060	8,060	8,060	8,060	8,060	8,060	8,060
Lubricants	400	400	400	400	200	200	400	400
Miscellaneous supplies	350	350	350	350	300	300	350	350
Repairs	1,858	1,870	1,600	1,645	1,009	1,090	1,611	1,654
Cost pumping cooling water	.....	.....	.....	.....	.....	.....	.....	.....
Total operating cost	$ 40,861	$41,105	$ 42,615	$ 42,427	$ 43,871	$ 44,779	$ 45,693	$ 45,803
Cost per kw.-hr. operation (cents)	2.74	2.74	3.05	2.98	3 48	3 49	3 26	3 21
Cost per kw.-hr. fixed charges (units)	1.93	1.95	1.82	1.86	1.52	1.62	1 96	2 03
Cost per kw.-hr. total (cents)	4.67	4.69	4 87	4.84	5 00	5.11	5.22	5 24
Cost per kw.-peak (dollars)	562.77	570.75	530.52	542.67	444 25	474.25	572.40	591.25
	B—STEAM EQUIPMENT DESIGNED							
INVESTMENT:								
Real estate	$ 2,500	$ 2,500	$ 2,500	$ 2,500	$ 2,500	$ 2,500	$ 2,500	$ 2,500
Brick building	35,200	35,800	35,200	35,800	26,000	28,000	43,000	45,000
Generating units, delivered and erected	31,954	32,250	25,504	26,634	16,750	19,449	23,380	24,450
Switchboards and street lighting transformers	6,500	7,000	6,500	7,000	6,500	7,000	6,500	7,000
Electric wiring and ducts	3,500	4,000	3,500	4,000	3,500	4,000	3,500	4,000
Piping complete	7,000	7,000	7,000	7,000	7,000	7,000	7,000	7,000
Condensing equipment	.....	.....	.....	.....	.....	.....	.....	.....
Foundations, exclusive of building	4,200	3,900	4,200	3,900	2,500	2,800	4,500	4,200
Oil filters and tanks	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800
Railroad siding	3,200	3,200	3,200	3,200	3,200	3,200	3,200	3,200
Boilers, delivered and bricked in	15,100	15,100	15,100	15,100	18,200	18,200	18,200	18,200
Feed water heater	900	900	900	900	900	900	900	900
Feed pumps	1,500	1,500	1,500	1,500	1,500	1,500	1,500	1,500
Steel stack and flues	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800
Motor-driven pump for cooling water	.....	.....	.....	.....	.....	.....	.....	.....
10,000 gal. fuel oil storage tank	.....	.....	.....	.....	.....	.....	.....	.....
Air compressor and tanks	.....	.....	.....	.....	.....	.....	.....	.....
Intake for circulating water	.....	.....	.....	.....	.....	.....	.....	.....
Total investment	$115,154	$116,750	$108,704	$111,134	$ 92,150	$ 98,149	$117,780	$121,550
COST OF OPERATION:								
Fixed charges 15 per cent. on investment	$ 17,273	$ 17,513	$ 16,306	$ 16,670	$ 13,822	$ 14,722	$ 17,667	$ 18,232
Fuel	12,725	12,660	15,400	14,820	20,300	19,900	17,860	17,300
Labor and superintendent	8,060	8,060	8,060	8,060	8,060	8,060	8,060	8,060
Lubricants	400	400	400	400	200	200	400	400
Miscellaneous supplies	350	350	350	350	300	300	350	350
Repairs	1,942	1,954	1,684	1,729	1,093	1,174	1,723	1,766
Cost pumping cooling water	.....	.....	.....	.....	.....	.....	.....	.....
Total operation cost	$ 40,750	$ 40,937	$ 42,200	$ 42,029	$ 43,775	$ 44,356	$ 46,060	$ 46,108
Cost per kw.-hr. operation (cents)	2.68	2.68	2.96	2.89	3.42	3.38	3.24	3.18
Cost per kw.-hr. fixed charges (cents)	1.97	2.00	1.86	1.90	1.58	1.68	2.02	2.08
Cost per kw.-hr. total (cents)	4.65	4.68	4.82	4.79	5.00	5.06	5.26	5.26
Cost per kw.-peak (dollars)	575.77	583.75	543.52	555.67	460.75	490.74	588.90	607.75

COST OF A 200 Kw. CENTRAL STATION
Units Except Corliss Engines. Steam Pressure 150 Lb. per Sq. In. Gage for Corliss
Ton Delivered. Cost of 18,500 B.T.U. Oil $3.00 Per Bbl. Delivered

Condensing steam prime movers Engines condensing to 26-in. vacuum, turbines condensing to 28-in. vacuum								Oil engines			
Uniflow engines		Counter-flow engines		Turbines		Corliss engines belted		Semi-diesel		Diesel	
2–200 kw.	3–100 kw.	2–200 kw.	3–100 kw.	2–200 kw.	3–100 kw.	2–200 kw.	3–100 kw.	2–200 kw.	3–100 kw.	2–200 kw.	3–100 kw.
FOR SATURATED STEAM											
2,500	2,500	2,500	2,500	2,500	2,500	2,500	2,500	2,500	2,500	2,500	2,500
38,000	38,800	38,000	38,600	28,800	30,800	45,800	47,800	26,400	30,100	26,400	23,500
31,954	32,250	25,504	26,634	19,460	23,100	23,380	24,450	56,556	48,220	85,600	72,500
6,500	7,000	6,500	7,000	6,500	7,000	6,500	7,000	6,500	7,000	6,500	7,000
3,500	4,000	3,500	4,000	3,500	4,000	3,500	4,000	3,500	4,000	3,500	4,000
9,500	10,500	9,500	10,500	9,500	10,500	9,500	10,500	2,700	3,000	2,700	3,000
7,000	10,500	7,000	10,500	7,000	10,500	7,000	10,500				
4,700	4,400	4,700	4,400	2,800	3,100	5,000	4,700	5,500	6,000	4,000	4,500
1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800
3,200	3,200	3,200	3,200	3,200	3,200	3,200	3,200	1,500	1,500	1,500	1,500
13,000	13,000	13,000	13,000	13,000	13,000	15,400	15,400				
900	900	900	900	900	900	900	900				
1,500	1,500	1,500	1,500	1,500	1,500	1,500	1,500				
1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800				
.....	.....	.....	.....	.....	.....	.....	.....	800	800	800	800
.....	.....	.....	.....	.....	.....	.....	.....	2,800	2,800	2,800	2,800
.....	.....	.....	.....	.....	.....	.....	.....	2,500	2,500	2,500	2,500
.....	.....	.....	.....	.....	.....	.....	.....	1,200	1,200	1,200	1,200
$125,854	$131,950	$119,404	$126,334	$102,260	$113,700	$127,780	$136,050	$114,256	$111,420	$141,800	$127,600
18,878	19,793	17,910	18,950	15,339	17,055	19,167	20,407	17,138	16,713	21,270	19,140
11,650	11,900	15,500	15,100	13,100	13,600	17,200	16,700	10,015	10,015	7,500	7,500
8,060	8,060	8,060	8,060	8,060	8,060	8,060	8,060	7,700	7,700	7,700	7,700
400	400	400	400	200	200	400	400	438	602	438	602
350	350	350	350	300	300	350	350	657	904	657	904
2,138	2,290	1,882	2,065	1,229	1,443	1,891	2,074	2,262	1,929	3,424	2,928
.....	.....	.....	.....	.....	.....	.....	.....	197	197	197	197
$ 41,476	$ 42,793	$ 44,102	$ 44,925	$ 38,228	$ 40,658	$ 47,068	$ 47,991	$ 38,407	$ 38,060	$ 41,186	$ 38,971
2.58	2.62	2.99	2.96	2.62	2.70	3.19	3.15	2.43	2.43	2.27	2.26
2.15	2.26	2.05	2.16	1.75	1.94	2.19	2.33	1.96	1.91	2.44	2.18
4.73	4.88	5.04	5.12	4.37	4.64	5.38	5.48	4.39	4.34	4.71	4.44
629.27	659.75	597.02	631.67	511.30	568.50	638.90	680.25	571.28	557.10	709.00	638.00
FOR STEAM SUPERHEATED 100° F.											
$ 2,500	$ 2,500	$ 2,500	$ 2,500	$ 2,500	$ 2,500	$ 2,500	$ 2,500				
38,000	38,600	38,000	38,600	28,800	30,800	45,800	47,800				
31,954	32,250	25,504	26,634	19,460	23,100	23,380	24,450				
6,500	7,000	6,500	7,000	6,500	7,000	6,500	7,000				
3,500	4,000	3,500	4,000	3,500	4,000	3,500	4,000				
10,000	11,000	10,000	11,000	10,000	11,000	10,000	11,000				
7,000	10,500	7,000	10,500	7,000	10,500	7,000	10,500				
4,700	4,400	4,700	4,400	2,800	3,100	5,000	4,700				
1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800				
3,200	3,200	3,200	3,200	3,200	3,200	3,200	3,200				
15,100	15,100	15,100	15,100	15,100	15,100	18,200	18,200				
900	900	900	900	900	900	900	900				
1,500	1,500	1,500	1,500	1,500	1,500	1,500	1,500				
1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800				
.....	.....	.....	.....	.....	.....	.....	.....				
.....	.....	.....	.....	.....	.....	.....	.....				
.....	.....	.....	.....	.....	.....	.....	.....				
.....	.....	.....	.....	.....	.....	.....	.....				
$128,454	$134,550	$122,004	$128,934	$104,860	$116,300	$131,080	$139,350				
$ 19,268	$ 20,183	$ 18,300	$ 19,340	$ 15,729	$ 17,445	$ 19,662	$ 20,902				
11,100	11,500	14,600	14,700	13,150	14,000	17,350	16,700				
8,060	8,060	8,060	8,060	8,060	8,060	8,060	8,060				
400	400	400	400	200	200	400	400				
350	350	350	350	300	300	350	350				
2,222	2,374	1,964	2,149	1,292	1,506	2,003	2,186				
$ 41,400	$ 42,867	$ 43,674	$ 44,999	$ 38,731	$ 41,511	$ 47,825	$ 48,598				
2.52	2.59	2.89	2.93	2.63	2.75	3.22	3.15				
2.20	2.30	2.09	2.21	1.79	1.99	2.24	2.39				
4.72	4.89	4.98	5.14	4.42	4.74	5.46	5.54				
642.27	672.75	610.02	644.67	524.30	581.50	655.40	696.75				

Thus, for the 200-kw. condensing-turbine plant, *the unit operating cost* = \$38,228 ÷ (365 × 2,400) = \$0.0437 or 4.37 *ct. per kw.-hr.;* see Table IV.

For comparison, the investment cost per kilowatt of peak load was also computed for each type of unit. Thus, for the 200-kw. condensing turbine plant, *the cost per kilowatt of peak load* = (*total investment cost*) ÷ (*kilowatts peak-load capacity*) = \$102,260 ÷ 200 = \$511.30.

6. Tabulate The Estimates And Decide On The Type Of Equipment.—The estimates are here tabulated in Table IV.

From the preceding tabulation (Table IV) it is evident that the plant with three 100-kw. semi-Diesel engines shows the least unit energy cost (4.34 ct. per kw.-hr.) which is but slightly less than that of the plant with two 200-kw. condensing turbines when supplied with saturated steam (4.37 ct. per kw.-hr.). Because of the lesser investment cost of the turbine plant and because of its greater reliability, it would probably, for the stated conditions, be chosen in preference to the oil-engine plant.

"Contrary to what seems to be the general belief, *the lower steam rate which obtains with superheated steam is, in practically all cases, offset by higher fixed charges* and fuel costs; and, except in the case of turbines, no net gain is realized by operating the plants condensing."

293. The Information Which Should Be Given The Turbine Manufacturer When Requesting A Quotation is as follows: (1) *What is wanted;* turbine, turbo-generator, turbine-driven centrifugal pumps, etc. (2) *Capacity;* horsepower, kilovolt-amperes, kilowatts, or gallons per minute; always, if possible, for an alternating-current generator, state the power factor. (3) *Speed;* this need not generally be given if the driven machine is to be included in the quotation. (4) *Steam conditions;* boiler pressure, superheat, and back-pressure or vacuum. If a mixed-pressure or bleeder turbine is wanted give also the quantity and pressure of the low-pressure steam which is available or to be extracted. (5) *Output conditions;* whether alternating-current or direct-current generator is wanted, voltage, number of phases and frequency or head against which pump must discharge, etc. If an a.-c. generator is required state whether the exciter is wanted direct-connected on the main-turbine shaft or whether separate turbine-driven exciter is wanted. (6) *Nature of load* on driven-machine or on turbine; state whether load is composed largely of motors or whether it is principally a lighting load and also whether the load is steady or variable.

ALLIS-CHALMERS MANUFACTURING COMPANY

PROPOSAL

Milwaukee, Wisconsin, U. S. A. June 1 1922

To Brown and Black Manufacturing Company
(Hereinafter called the Purchaser)

Address St. Louis, Mo.

Allis-Chalmers Manufacturing Company, hereinafter called the Company, proposes to furnish the Purchaser, on the following conditions, the machinery described below, or in the Company's specifications attached, which are made a part of this proposal, f. o. b. cars point of shipment.

One (1) 750-kw., at 80 per cent power factor, maximum rated turbo-alternator unit complete with expansion joint but not including exciter, as per attached specifications pages 5 to 9 inclusive.

All machinery shall be installed by and at the expense of the Purchaser, unless otherwise expressly stipulated herein.

The Company will repair f. o. b. works where made, or furnish without charge f. o. b. its works, a similar part to replace any material of its own manufacture which, within one year after shipment, is proven to have been defective at the time it was shipped, provided the Purchaser gives the Company immediate written notice of such alleged defects. The Company shall not be held liable for any damages or delays caused by defective material, and no allowance will be made for repairs or alterations, unless made with its written consent or approval.

The title and right of possession to the machinery herein specified, remains in the Company until all payments hereunder, (including deferred payments and any notes or renewals thereof, if any), shall have been fully made in cash, and it is agreed that the said machinery shall remain the personal property of the Company whatever may be the mode of its attachment to realty or otherwise, until fully paid for in cash. Upon failure to make payments, or any of them, as herein specified, the Company may retain any and all partial payments which have been made, as liquidated damages, and shall be entitled to take immediate possession of said property, and be free to enter the premises where said machinery may be located, and to remove the same as its property without prejudice to any further claims on account of damage which the Company may suffer from any cause. The company may pursue all legal remedies to enforce payment hereunder, but if unable to collect may thereafter repossess the property.

The Company agrees that it shall at its own expense defend any suits that may be instituted by any party against the Purchaser, for alleged infringement of patents relating to machinery of its own manufacture furnished under this proposal, provided such alleged infringement shall consist in the use of said machinery, or parts thereof, in the regular course of the Purchaser's business, and provided the Purchaser shall have made all payments then due under this contract, and gives to the Company immediate notice in writing of the institution of such suits, and permits the Company, through its Counsel, to defend the same, and gives all needed information, assistance and authority to enable the Company to do so, and thereupon in case of a final award of damages in such suit the Company will pay such award, but it shall not be responsible for any compromise made without its written consent, nor shall it be bound to defend any suit or to pay any damages therein when the same shall arise by reason of the use of parts not furnished by the Company under this proposal. The Company shall also be notified of, and reserves the right to be represented at any tests which the Purchaser may make, in relation to guarantees of operation.

If shipment of the machinery herein specified, or any part thereof, is delayed by any cause for which the Company is not directly or indirectly responsible, the date of completion of said machinery by the Company shall be regarded as the date of shipment in determining when payments for said machinery are to be made, and the Company shall be entitled to receive reasonable compensation for storage; such storage to be at the risk of the Purchaser. If all the machinery should not be forwarded on the same date, pro-rata payments shall be made for partial shipments. All notes and securities given to the Company by Purchaser are taken by the Company, not in payment, but as evidence only of Purchaser's indebtedness.

This contract is contingent upon strikes, fires, accidents or other delays unavoidable or beyond the reasonable control of the Company. The Company shall not be held responsible or liable for any loss, damage, detention or delay, from any cause beyond its control; and the receipt of the machinery by the Purchaser shall constitute acceptance of its delivery and a waiver of any and all claims for loss or damage due to any delay.

FIG. 271.—Typical manufacturer's proposal (part I; this constitutes pages 1 and 2 of this particular proposal).

PRICE:—The price of said machinery is .....

..... Twenty Eight Thousand *Dollars,*

($ 28,000.00 ....), *payable in New York, Chicago or Milwaukee Exchange*

TERMS.—Terms of payment are as follows:

60 °/o Cash upon presentation of Bill of Lading

20 °/o Cash 30 days thereafter

20 °/o Cash 60 days thereafter

SHIPMENT:—The machinery herein specified will be shipped 120 days

..... *from the date of the receipt of the contract and final information from the Purchaser, at the Company's works.*

The services of engineers, millwrights or mechanics furnished by the Company for the purpose of superintending the erection or operation of the machinery covered by this proposal, shall be paid for by the Purchaser, monthly and independent of the contract account, at the rate of Fifteen Dollars per eight hour day and regular overtime rates plus all traveling and hotel expenses, including all time the said parties are absent from the Company's works on the Purchaser's business; it being understood and agreed that during the term of such service the said engineers, millwrights and mechanics shall be the Purchaser's employees, for whose acts the Company shall assume no responsibility. All labor and material required in connection with these services, will be furnished by the Purchaser.

In the event it is elsewhere herein agreed that the Company shall erect the machinery herein specified, the Purchaser shall reimburse the Company for all expenses in connection with the erection of the machinery occasioned by delays, lack of facilities or apparatus to be furnished by the Purchaser or any acts for which the Company is not responsible.

In the event the Company furnishes oil, wire, cable or other material requiring special carriers, under this proposal, (such as oil barrels, reels, etc.), the Purchaser will pay to the Company, under the terms of this agreement, the value of such carriers in addition to the contract price. Upon return of such carriers, in good condition, to the proper receiving point, to be designated by the Company, the Company will credit the Purchaser the full amount previously charged; provided, however, that invoice or memorandum and necessary shipping documents are promptly forwarded to the Company and return shipment is made, charges prepaid, within four months from the original date of shipment.

The Purchaser shall provide and maintain in the name of the Company adequate insurance for the machinery herein specified, against loss or damage by fire, in an amount fully protecting the Company, and the Purchaser shall assume all loss resulting from fire in case of failure to effect such insurance.

All the terms and provisions of the contract between the parties hereto, are fully set out herein, and no agent, salesman or other party is authorized to bind the Company by any agreement, warranty, statement, promise or understanding not herein expressed, and no modifications of the contract shall be binding on either party unless the same are in writing, accepted by the Purchaser and approved in writing by an Executive Officer of the Company.

This proposal is for immediate acceptance of the Purchaser, and is subject to the written approval of an Executive Officer of the Company, and shall not be binding upon the Company until so approved.

ALLIS-CHALMERS MANUFACTURING COMPANY,

By.....

ACCEPTANCE.

The foregoing proposal is hereby accepted

this.....day of .....192...

(Purchaser Sign Here) .....

Fig. 272.—Typical manufacturer's proposal (part II; this constitutus pages 3 and 4 of this particular proposal).

ALLIS-CHALMERS MANUFACTURING COMPANY

MILWAUKEE, WISCONSIN, U. S. A.

MIXED PRESSURE CONDENSING STEAM TURBINE AND ALTERNATOR UNIT

For Brown and Black Manufacturing Company, St. Louis, Mo.

These guarantees and specifications form part of proposal dated June 1, 1922.

STEAM CONSUMPTIONS

The steam turbine unit described in the following pages, when erected and properly adjusted in the Purchaser's power house, will carry true energy steady loads as given below at 80 per cent power factor and under constant operating conditions as set forth on page 6 of these specifications, with a consumption of dry steam not exceeding:—

		Low Pressure Condensing	High Pressure Condensing
At One-half load	(viz.: 375 K. W.)	41.6 lbs. per K. W. hour	21.2 lbs. per K. W. hour
At Three-quarters load	(viz.: 562.5 K. W.)	36.6 lbs. per K. W. hour	18.8 lbs. per K. W. hour
At Full load	(viz.: 750 K. W.)	34.3 lbs. per K. W. hour	17.5 lbs. per K. W. hour

The above steam consumptions do not include steam or power used by accessories and auxiliaries. The above loads are the true electrical output at the generator terminals and include energy required in the field for excitation. When the proposal includes an exciter direct connected to this steam turbine unit the steam required to drive same is included. When steam turbine is operating on low pressure steam provision will be made for admitting a small amount of high pressure steam to keep high pressure blading cool.

Rated capacity of unit at 80 per cent power factor 750 K. W. Maximum.

Rated current per terminal 225 Amperes.

Normal Voltage 2300, Cycles 60, Phase 3

Normal speed 3600 revolutions per minute.

Turbine to be operated condensing.

Steam pressure at turbine high pressure throttle 200 pounds by gauge.

Steam pressure at turbine low pressure throttle 16 pounds absolute.

Superheat in steam at turbine high pressure throttle 0 degrees Fahrenheit.

Superheat in steam at turbine low pressure throttle 0 degrees Fahrenheit.

Vacuum at turbine exhaust nozzle 28 inches of mercury (Referred to 30" Barometer)
(Back pressure at exhaust nozzle not to exceed 3 lbs. by gauge when operating temporarily noncondensing with high pressure steam.)

The maximum temperature of the generator, with 40°C. inlet air temprature, will not exceed:—

K. W.	Power Factor %	K. V. A.	Amperes per terminal	Hours	Stator °C.	Rotor °C.
750	80	937.5	235	24	100	135

Excitation voltage 125 Approximately 55 amperes excitation will be required with rated current at one hundred per cent power factor. Approximately 70 amperes excitation will be required with the same current at eight per cent power factor.

Insulation test, a. c. for one minute: Field 1500 volts; Armature 5600 volts.

Diameter of H. P. steam inlet at turbine 5 inches

Diameter of L. P. steam inlet at turbine 14 inches. Turbine exhaust nozzle 30 inches.

Approximate overall length of unit above floor 18 feet, 11 inches.

Approximate additional length below floor 3 feet, 0 inches.

Approximate overall width of unit 6 feet, 11 inches.

*Approx. height of highest point of unit, above floor line 9 feet, 0 inches.

Approximate shipping weight of unit 46500 pounds.

Approximate weight of heaviest piece to be handled in erecting 25360 pounds.

Approximate weight of heaviest piece to be handled after erection 3120 pounds.

Approximate amount of air required by generator per minute 4850 cu. ft.

*(NOTE—Items marked thus * is the height, if the unit is set as designed. It sometimes happens that a unit is set higher on account of local conditions, in which case this figure will be increased.)
(NOTE—If more room is necessary add extra sheets giving them this page number with subletters. Items inserted take precedence over the printed specifications in case of conflict.)

FIG. 273.—Typical manufacturer's proposal (part III; this constitutes pages 5 and 6 of this particular proposal).

MIXED PRESSURE CONDENSING STEAM TURBINE AND ALTERATOR UNIT.

GENERAL DESCRIPTION—The steam turbine will be of the horizontal, Allis-Chalmers reaction type, connected to the generator by a flexible coupling. The rotors of turbine and generator will each be carried in two bearings, so that either rotor may be handled separately.

BLADING—The blading will be of the Company's patented construction, made of materials especially adapted to resist corrosion, erosion and steam temperature specified.

GLANDS—An adequate supply of clear water, free from acid or scale forming impurities, and at a steady pressure of fifteen pounds by gauge at the glands, shall be furnished by the Purchaser.

GOVERNOR—The governor will be provided with a hand operated synchronizer, arranged so that the mean speed of the unit may be varied approximately three per cent above or below the normal.

REGULATION—The variation in speed from half load to full load under ordinary operating conditions, will be approximately three per cent; great or sudden variations of load may cause approximately five per cent momentary speed variation.

SAFETY STOP—A separate safety stop governor will be supplied, which will automatically shut off the steam if unit reaches a predetermined speed in excess of the normal. A lever for tripping safety stop by hand is conveniently located on unit.

THROTTLE VALVE—Screw-operated high pressure and low pressure throttle valves will be provided. Unit will be arranged so that high pressure steam will be automatically admitted in case low pressure steam supply is not sufficient to carry the load. Unit may be operated entirely with low pressure steam or entirely with high pressure steam.

GAUGES—The Company's standard equipment of gauges and gauge board will be provided on unit.

HAND OF TURBINE—The turbine will be according to the Company's standard practice.

ARMATURE—The armature core will be built up of laminated steel held in slots in the cast iron frame. Ventilating spaces will be provided through which air will be forced. The coils, thoroughly insulated, will be firmly held in slots in the laminated core. A supply of clean cool air for generator shall be arranged for and supplied by Purchaser.

FIELD—The core of the revolving field will be made of steel with slots to receive the windings. The windings will be of copper securely held in the slots by wedges. The ends of the coils will be substantially supported. The alternator will be ventilated by air forced through all parts by means of fans attached to the field.

EXCITER—The exciter is not included unless so specified. Connections to same are not included. When exciter is included Purchaser shall promptly advise winding desired for same.

RHEOSTAT—The alternator will be provided with a field regulating rheostat, arranged for installation behind the switchboard; rheostat includes face-plate and means for operating by hand from front of switchboard.

INSTRUMENTS—Instruments, such as ammeters, voltmeters, synchronizing switch, etc., are not included under these specifications.

TERMINALS—No terminals for armature leads are included, these leads will be arranged for soldering to the cables leading to the switchboard. No cables or wiring is included.

PARALLEL OPERATION—This turbo-generator unit will operate in parallel with similar units; also with other units which fulfill the requirements for parallel operation, and have a speed regulation similar to that of this unit.

LUBRICATION—A self-contained oiling system will be supplied. The Purchaser shall furnish adequate clear cool water, free from acid or scale-forming impurities for oil cooler. The Purchaser shall provide lubricating oil of proper quality and suitable character.

PAINTING—All exposed unfinished parts will be painted with one coat of black paint before shipping. No ornamental painting or painting after shipment is included.

TOOLS—The Company's standard equipment of wrenches and tools will be furnished. When more than one turbine is included in the contract, only one set of wrenches and tools will be furnished.

FOUNDATIONS—The Purchaser shall provide suitable foundations, including material and labor for grouting under the unit after same has been lined up and leveled by Company's engineer, also such sub-foundations, air cleanser and air ducts, for which the Company does not furnish drawings, as the local conditions necessitate. The foundations and sub-foundations must be so constructed that they will not receive or transmit vibrations from or to the adjacent flooring or structure. The Company will furnish its standard outline and foundation plan drawings of apparatus furnished under these specifications. Purchaser shall furnish drawings of foundations, air ducts, etc., and shall submit same to Company before any work is done. The Purchaser shall furnish foundation template and foundation bolts and washers.

PIPING—The Purchaser shall furnish all steam and exhaust piping of ample size to give contract conditions at turbine and shall arrange same so that no strains or vibrations will be transmitted to the turbine. The exhaust pipe must be securely anchored under exhaust nozzle, laid out to avoid a stiff construction; and must be provided with a suitable expansion joint at the turbine exhaust nozzle. The Purchaser shall provide suitable size exhaust free to atmosphere provided with a water sealed automatic relief valve, if a gate valve is located in the turbine exhaust line, this atmospheric connection must be placed on the turbine side of same. Purchaser shall provide proper relief valve in low pressure line to turbine also proper drains and traps for all piping and shall furnish an efficient steam and oil separator near turbine L. P. throttle; also an efficient steam separator near turbine H. P. throttle. The arrangement of all steam and exhaust piping shall be submitted by the Purchaser to the Company before any work is done. Purchaser shall furnish all water piping to and from unit.

OPERATION—The steam turbine unit will operate successfully after being properly erected and adjusted, provided it receives such care and attention as is necessary and usual for units of this type and size; this includes the proper operation of the condenser and of the boiler plant, avoiding slugs of water and unduly wet steam also great or sudden fluctuations of temperature or pressure. It is understood that the usual operating conditions will be as specified herein.

Fig. 274.—Typical manufacturer's proposal (part IV; this constitutes pages 7 and 8 of this particular proposal).

294. Turbine Specifications And Guarantees (Figs. 273, 274 and 275) are sent with the manufacturer's proposal (Figs. 271, and 272) and form a part of the proposal. Although the proposal which is here shown is for a mixed-pressure turbo-alternator unit, it is typical of those furnished for all classes of turbines. The proposal, when accepted and signed, by both

MIXED PRESSURE CONDENSING STEAM TURBINE AND ALTERATOR UNIT.

PACKING FOR SHIPMENT—The turbine and generator will be prepared in the Company's usual manner for domestic rail shipment. Packing for foreign shipment or for coastwise shipment is not included unless so specified.

IN GENERAL—These specifications cover the Company's standard turbine-generator unit with standard equipment complete as described, beginning at the inlet flange of the throttle valves, and ending at the flange of the exhaust nozzle and at the generator terminals. It is advisable that the Purchaser provide the Company promptly with drawings of the power house in the vicinity of the turbine location, showing other machinery, columns and foundations, existing and proposed piping, proposed arrangement of condensing apparatus, etc. Purchaser shall provide proper space for installing unit and for removal of generator rotor.

TESTS—When tests are desired to determine that the unit fulfills the guarantees set forth in these specifications same will be made at Purchaser's plant by and at the expense of Purchaser, and within thirty days after unit is ready for Purchaser's connections. The Purchaser shall give the Company two weeks' written notice of his intention to make tests, and shall permit the Company at its expense to have representatives in the power plant prior to and during tests; and to furnish and couple up such instruments as the Company may desire. Prior to the tests the Company shall have reasonable access to the unit for examination and the Purchaser shall make necessary preliminary tests. The conditions under which tests will be made, calibration of instruments, methods of measurements, etc., shall be mutually agreed upon between the Purchaser and the Company; in general the rules of the A. S. M. E. and the A. I. E. E. will be followed. Insulation tests will be made according to the rules of the A. I. E. E.

GENERATOR TEMPERATURES—Generator temperatures will be measured in accordance with the Standardization Rules of the A. I. E. E. as follows: Stator: For units 500 KVA or smaller, by thermometer applied to the hottest accessible part of the completed machine; to the temperature so determined will be added 15°C. correction. For units over 500 KVA the temperature will be measured by embedded resistance coils placed as nearly as possible at the hottest part of the winding; to this temperature will be added 5°C. correction. Resistance temperature coils are included, but no instrument will be furnished. Rotor: will be measured by increase of resistance of the winding; to the temperature so determined will be added 10°C. correction

ERECTION—For the purpose of superintending the erection and starting of the machinery described herein, the Purchaser agrees to and will engage and pay for the services of such erecting engineers to be furnished by the Company as may be necessary, as provided in attached proposal. If, however, this proposal requires the Company to furnish engineers at its expense, the Purchaser shall place machinery in power house adjacent to turbine foundation and the erection of the machinery shall commence immediately upon engineer's arrival at Purchaser's plant and proceed to completion without delay. The turbine engineer will remain at the Purchaser's plant, for operation, not longer than one week after the machinery is erected; it being understood and agreed that Purchaser's part of the work will be completed when erection of steam turbine unit is complete. The Purchaser shall pay the Company for the time and expenses of the engineer beyond this period; also all time and expenses caused by delays which occur in the erection, starting, or operation of the machinery, provided the Company is not responsible for such delays. It is understood that the erecting engineers will not work more than ten hours per working day. Overtime and night work also work on Sundays and Legal holidays, must be especially arranged for between the Purchaser and the Company. The Purchaser shall give the Company at least one week's written notice of the date when he will be ready for the erecting engineer.

ALLIS-CHALMERS MANUFACTURING COMPANY,

By.....

FIG. 275.—Typical manufacturer's proposal (part V; this constitutes page 9 of this particular proposal).

the purchaser and manufacturer, forms a binding contract between the two. By the contract, the manufacturer can be held to the fulfillment of the specifications and the guarantees. If the turbine in an *acceptance test* (see Fig. 275) does not perform as well as is stipulated in the guarantee, the purchaser has the right to reject the machine or to receive a liberal reduction in the specified purchase price.

295. In Selecting The Best Steam Conditions Under Which To Operate A Contemplated Turbine, as must be done when

an entire plant is being designed, the unit operating cost (Sec. 271) is again the deciding factor. By computing the unit operating cost for various steam conditions, those conditions can be found which afford the least unit cost. Generally speaking, the operating costs of turbines decrease (see Div. 13) with higher initial pressures, higher superheats, and lower back pressures; but the operating costs of the boilers and condensers go up as those of the turbines go down. Hence, the selection of the best operating conditions is again a matter of economics and must be executed with a view toward attaining the minimum unit operating cost.

Note.—The Most Usual Steam Conditions In Turbine Plants are: (1) *For small plants* (up to about 1,000 kw.) initial pressures of 150 to 200 lb. per sq. in. gage, superheats up to about 125° F., and vacuums of 27.5 to 28 in. (2) *For large plants* initial pressures of 200 to 300 lb. per sq. in. gage, superheats up to about 200° F., and vacuums of 28.5 to 29 in. The tendency is toward the use of higher steam pressures; several plants have been built for 350 lb. boiler pressure.

QUESTIONS ON DIVISION 14

1. What are the three principal objects of studying turbine operating costs?

2. Enumerate eight factors which are usually considered as items of operating cost and arrange them into two groups. What are the names of the two groups?

3. What is the usual annual amount of the fixed charges for turbines? How is the amount determined in any given case?

4. Why are the fixed charges so called?

5. Explain the meaning of the term *unit charges*. *Unit operating cost.*

6. State as many factors as you can that affect the unit operating charges of a plant and show their effect.

7. Define *load factor* and show how it affects the unit operating costs and the annual operating and fixed charges.

8. What other operating costs are generally included with those of a turbine? Why?

9. What three factors must be considered when selecting the type of prime mover for a given service?

10. For what classes of service is the steam turbine best adapted? Why?

11. What classes of services are quite beyond the field of the steam turbine? Why?

12. State what you can regarding the reliability of steam turbines.

13. Upon what does the efficiency or steam economy of a turbine depend?

14. About what steam rates may be expected from each of the following-sized turbines when operating condensing and when operating non-condensing: 50-kw.? 200-kw.? 500-kw.? 1,000-kw.? 2,000-kw.? 3,500-kw.?

15. State how the efficiency of a turbine varies with the load which it delivers.

16. What is the meaning of a turbine's capacity rating?

17. What can you say of the economy, in dollars and cents, of low- and mixed-pressure turbines? Explain.

18. Is it advisable, usually, to employ separate high- and low-pressure turbines? Why?

19. Wherein does the economy of bleeder turbines lie? Explain.

20. How would you predict the steam rate of a contemplated turbine?

21. Upon what do the relative economies of steam turbines and steam engines depend?

22. State several advantages which, in general, the steam turbine has over the steam engine and *vice versa*.

23. What can you say, in general, of the relative steam economies of non-condensing engines and turbines?

24. In general, which has the better steam economy, a condensing engine or a condensing turbine?

25. State the principal services for which turbines and engines of small capacity are each adapted.

26. How do the prices of steam turbines vary with their capacities? Give some typical prices.

27. Enumerate the steps which should be taken in selecting a prime mover for a given service, explaining each step as fully as possible.

28. State briefly what information should be given to the turbine manufacturer when a quotation is requested.

29. What is the purpose of performance specifications and quarantees in steam-turbine proposals? How are they enforced?

30. How are the best steam conditions for a proposed turbine plant determined? What are the most usual steam conditions in practice?

SOLUTIONS TO PROBLEMS ON DIVISION 1

STEAM-TURBINE FUNDAMENTAL PRINCIPLES

1. From the total-heat-entropy chart of Fig. 15, $\mathbf{H}_1 = 1{,}210$ *B.t.u.* $\mathbf{H}_2 = 1{,}022$ *B.t.u.* Hence, *heat liberated* $= \mathbf{H}_1 - \mathbf{H}_2 = 1{,}210 - 1{,}022 = 188$ *B.t.u. per lb.*

2. From the total-heat-entropy chart of Fig. 15, $\mathbf{H}_1 = 1{,}189$ *B.t.u.* $\mathbf{H}_2 = 887$ *B.t.u.* Hence, *heat liberated* $= \mathbf{H}_1 - \mathbf{H}_2 = 1{,}189 - 887 = 302$ *B.t.u. per lb.*

3. By For. (2): $v = 223.7\sqrt{\mathbf{H}_1 - \mathbf{H}_2} = 223.7\sqrt{188} = 223.7 \times 13.7 = 3{,}065$ *ft. per sec.* Or $v = 3{,}065 \times 60 \div 5{,}280 = 34.8$ *mi. per min.* By charts *B* and *C*, Fig. 15, $v = 3{,}050$ *ft. per sec.*

4. By For. (17): $W = 778\mathbf{W}(\mathbf{H}_1 - \mathbf{H}_2) = 778 \times 1 \times 302 = 235{,}000$ *ft.-lb.*

5. By For. (18): $\mathbf{P} = \mathbf{W}(\mathbf{H}_1 - \mathbf{H}_2)/2{,}545 = 2{,}000 \times 302 \div 2{,}545 = 237$ *hp.* By For. (19): $\mathbf{W}_H = 2{,}545/(\mathbf{H}_1 - \mathbf{H}_2) = 2{,}545 \div 302 = 8.43$ *lb. per hp.-hr.* From *AB*, Fig. 15: $\mathbf{W}_H = 8.4$ *lb. per hp.-hr.*

6. From Fig. 20, for a 2,000-hp. turbine: $\mathbf{E}_r = 65$ per cent $= 0.65$. Hence, by For. (26): $\mathbf{W}_B = 2{,}545/[\mathbf{E}_r \times (\mathbf{H}_1 - \mathbf{H}_2)] = 2{,}545 \div [0.65 \times 188] = 20.8$ *lb. per b.hp-hr.* Hence, at full load, $\mathbf{W} = \mathbf{P}_B \times \mathbf{W}_B = 2{,}000 \times 20.8 = 41{,}600$ *lb. per hr.*

7. By Sec. 16, for maximum work: $v_b = v_i/2 = 3{,}065 \div 2 = 1{,}532$ *ft. per sec.*

SOLUTIONS TO PROBLEMS ON DIVISION 8

REGENERATORS AND CONDENSERS

1. By For. (28): $\mathbf{W}_{W1} = t\mathbf{W}_{S1}(\mathbf{L}_1 + \mathbf{L}_2)/2(T_1 - T_2)$. Now from steam tables, $T_1 = 240.1°$ F. $T_2 = 233.1°$ F. $\mathbf{L}_1 = 952$. $\mathbf{L}_2 = 956.7$.

Hence, $W_{W1} = 3 \times (1{,}500 \times 25 \div 60) \times (952 + 956.7) \div [2 \times (240.1 - 233.1)] = 3 \times 625 \times 1{,}908.7 \div 14 = 262{,}772$ *lb.* Also, by For. (29): $W_{W2} = W_{S2}(L_1 + L_2)/2(T_1 - T_2) = 2{,}000 \times (952 + 956.7) \div [2 \times (240.1 - 233.1)] = 2{,}000 \times 1{,}908.7 \div 14 = 272{,}671$ *lb.*

2. The condenser must handle $1{,}500 \times 25 = 37{,}500$ *lb. of steam per hr.* By Sec. 173, the steam temperature should be at least $60 + 25 = 85°$ F. From Fig. 184, the temperature at 1.5 in. pressure is 92° F. Hence, it is feasible to operate with this condenser pressure. The discharge circulating water temperature should not exceed $92 - 10 = 82°$ F. Assume a 20° F. rise through the condenser. The rate of heat transfer with this pressure may be assumed at 350 B.t.u. per sq. ft. per hr. per degree difference. Hence, using Fig. 184, and beginning at 1.5 in. pressure on the lower scale and following upward to the 60° F. line, to the left to the 20° F. rise line, upward to the 350 B.t.u. line, to the left to the curve and upward to the surface scale, there results a value of 125 *sq. ft.* per 1,000 lb. steam. The *condenser surface* $= 37.5 \times 125 = 4{,}687.5$ *sq. ft.* The *circulating water required* $= 37.5 \times 95 = 3{,}562.5$ *gal. per min.*

SOLUTIONS TO PROBLEMS ON DIVISION 9

HIGH-PRESSURE, BLEEDER, MIXED-PRESSURE, AND EXHAUST-STEAM TURBINES

1. By Fig. 20, *the efficiency ratio* $= 0.55 = E_r$. *The low-pressure steam rate* $= W_{B2} = 6{,}000 \div 500 = 12$ *lb. per hp.-hr.* From the total-heat-entropy chart of Fig. 15, $H_1 = 1{,}225$ B.t.u. per lb. $H_2 = 877$. $H_1' = 1{,}156$. $H_2' = 952$. Hence, by For. (30):

$$W_{B1} = \frac{1}{H_1 - H_2}\left[\frac{2{,}545}{E_r} - W_{B2}(H_1' - H_2')\right] = \left[1 \div (1{,}225 - 877)\right] \times \{(2{,}545 \div 0.55) - [12 \times (1{,}156 - 952)]\} = (1 \div 348) \times [4{,}630 - (12 \times 204)] = (4{,}630 - 2{,}448) \div 348 = 2{,}182 \div 348 = 6.3 \text{ lb. per hp.-hr.}$$

Hence, *total high-pressure steam required* $= 500 \times 6.3 = 3{,}150$ *lb. per hr.*

2. By Fig. 20, *the efficiency ratio* $= E_r = 0.63$. *The extraction rate* $= W_{B2} = 22{,}500 \div 1{,}500 = 15$ *lb. per hp.-hr.* From Fig. 15, $H_1 = 1{,}257$ B.t.u. per lb. $H_2 = 868$. $H_1' = 1{,}257$. $H_2' = 1{,}091$. Hence, by For. (30):

$$W_{B1} = \frac{1}{H_1 - H_2}\left[\frac{2{,}545}{E_r} - W_{B2}(H_1' - H_2')\right] = \left[1 \div (1{,}257 - 868)\right] \times \{(2{,}545 \div 0.63) - [15 \times (1{,}257 - 1{,}091)]\} = (1 \div 389) \times [4{,}040 - (15 \times 166)] = (4{,}040 - 2{,}490) \div 389 = 1{,}550 \div 389 = 3.99 \text{ lb. per hp.-hr.}$$

Hence, *total steam required* $= 1{,}500 \times 3.99 + 22{,}500 = 5{,}980 + 22{,}500 = 28{,}480$ *lb. per hr.*

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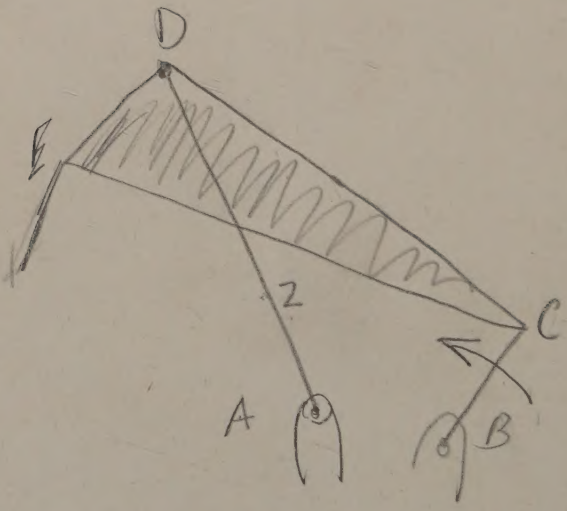

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